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MARINE ENGINEERING

(A TEXT-BOOK)

BY

ENGINEER-COMMANDER A. E. TOMPKINS

ROYAL NAVY

LATE INSTRUCTOR IN STEAM AND MARINE ENGINEERING, MACHINE CONSTRUCTION, ETC. AT THE ROYAL NAVAL COLLEGE, GREENWICH, AND LECTURER AT THE ROYAL NAVAL WAR COLLEGE; AUTHOR OF 'TURBINES,' ETC. MEMBER OF THE ROYAL SOCIETY OF ARTS, ETC.

THIRD EDITION

ENTIRELY REWRITTEN, REVISED, AND ENLARGED

WITH ABOUT 400 ILLUSTRATIONS

MACMILLAN AND CO., LIMITED

ST. MARTIN'S STREET, LONDON

1908

First Edition, 1896. *Additions*, 1898. *Additions*, 1900. *Second Edition, Enlarged and Rewritten*, 1904. *Additions*, 1906. *Third Edition, Enlarged and Rewritten*, 1908.

PREFACE TO THIRD EDITION

THE whole of the book has again been rewritten and revised, and, with a large number of new illustrations, forms a complete text-book on the construction and working of marine engines and boilers. The number of chapters has been increased from twenty-five to thirty-five, and the various chapters grouped into ten sections. The introductory section is developed so as to cover the syllabus used in the naval training establishments. Very little space is devoted to obsolescent types of boilers and machinery; the thermo-dynamics of the working substance is more fully developed so as to cover the ordinary requirements of the sea-going engineer; the auxiliary section includes new matter on steering and capstan gear, air compressors, hydraulic machinery, refrigerators, electrical and other machinery, as well as a chapter on water-tight, pumping, and fire systems; care and management comes in for a larger share of attention due to recent progress, and also includes a chapter on the duties of the Engineer of the Watch. In the last section, chapters are devoted to the marine steam turbine and to the internal combustion engine.

Every effort has been made to explain matters in as simple a manner as possible, but it should always be remembered that the practical application of mathematics is essential to the design and economical working of marine machinery. Having had nearly four years' experience as an instructor and lecturer in Marine Engineering,

192146

and many years' sea service in charge of machinery in warships of many types, the author hopes that the book will be accepted as a summary of the best practice of the present day.

Information has been derived from many sources as well as from some of my brother officers, to many of whom it has given me great pleasure to lecture, and from whom I have learnt much, both directly and indirectly. Many engineering firms have very kindly supplied me with details, matter, and sketches for illustrations, and in some cases I have been able to avail myself of the privilege of seeing machinery and boilers in course of construction. The source of information is acknowledged as far as possible; and the author also desires to acknowledge the kindness and courtesy of *Engineering*, *The Engineer*, *Power*, and other publications, in permitting the use of illustrations.

My best thanks are also given to Engineer-Lieutenant L. R. Croisdale for his careful reading of the second edition, and to Engineer-Lieutenants Leslie Robins and W. T. Parez for reading parts of the present edition.

A. E. TOMPKINS.

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PART I

CHAPTER I

THE MOTIVE POWER OF THE ENGINE, AND ITS DEVELOPMENT

Introductory.—In the ordinary steam engine, a force or pressure of steam acts on one side of a piston, or circular disc, and thus produces motion of a piston. This force overcomes another force opposed to it, which may be considered to act on the opposite side of the piston, and motion thus produced is applied to some useful purpose in doing work.

As shown in Fig. 1, suppose such a piston can be moved easily by hand, without allowing steam to leak past its edge, inside a cylinder which has its bottom end closed. Let the piston rest on a little water in the bottom of the cylinder, and then apply a fire so as to heat the water. At first the piston is pushed upwards as the water gets hotter, because the water expands slightly, but the distance the piston moves is not easily measured. If the piston be very light, and practically weightless, the water will begin to boil at a temperature of 212° Fahr.; and, steam being formed, the piston is pushed upwards because the steam thus produced fills 1643 times the volume or space that the water does from which it is formed. Thus, if all the water be changed into steam, the piston rises to a height 1643 times the original depth of the water.

FIG. 1.—Piston and Cylinder.

If the piston be very heavy, or loaded with a heavy weight, the water does not begin to boil until it is much hotter than before,

and the steam also fills less space or volume. Consequently, the piston rises to only a moderate height, as shown in Fig. 2; therefore it is seen that, when steam is formed, a large weight can be lifted, and this simply by heating water beneath a piston in a cylinder.

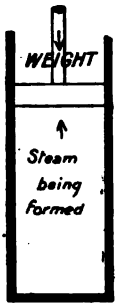


FIG. 2.

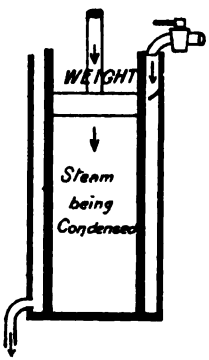


FIG. 3.

Next, suppose the piston and weight have been lifted, and that the fire is now removed, and that the cylinder is cooled by spraying cold water round it (as shown in Fig. 3); then the steam is condensed, and, filling less space or volume as water instead of steam, the piston falls back to its original position.

In an engine as above described, three separate operations are performed in the cylinder. Steam is generated, and the cylinder thus acts as a *boiler*; the piston is moved, and work is done in overcoming resistance, thus making the cylinder and piston a machine, or *engine*; and the steam is condensed to water, thus the cylinder acts as a *condenser*.

In 1698 Savery took out a patent "for raising water by means of the elastic force of steam." This engine was apparently never usefully employed, although, in a modified form, its action is very similar to that of the well-known pulsometer pump.

In 1712 Newcomen, who appears to have worked in conjunction with Savery, produced the first commercially useful engine, and it was erected at Dudley Castle. For the first time a separate vessel (*B* in Fig. 4) was used, and steam was generated in it as a *boiler*.

The figure shows the engine in its position of rest when not at work; the piston is at the top of its stroke in the cylinder *F*. On top of the standard *XX* is a crossbeam, *LKM*, pivoted at *K*. To the end *M* the piston, working in the cylinder *F*, is suspended by chains. To the other end *L* the plunger *P* and pump rod are similarly suspended; and to these is added a weight, *W*, so that there is a preponderance at this end *L*. Chocks of wood, *NN*, limit the vertical see-saw movement of the crossbeam about its pivot at *K*, and therefore also limit the *stroke*, or up-and-down movement, of the steam piston and pump plunger.

The fire is lighted under the boiler *B*, on the firegrate *C*. Steam, being generated in the boiler, is admitted at first by hand, but afterwards automatically, to the cylinder *F*, through the cock *A*. After completing the first downward stroke of the piston, the greater weight at the end *L* causes the piston to rise, and allows the steam to fill the space below the piston in the cylinder. When the piston

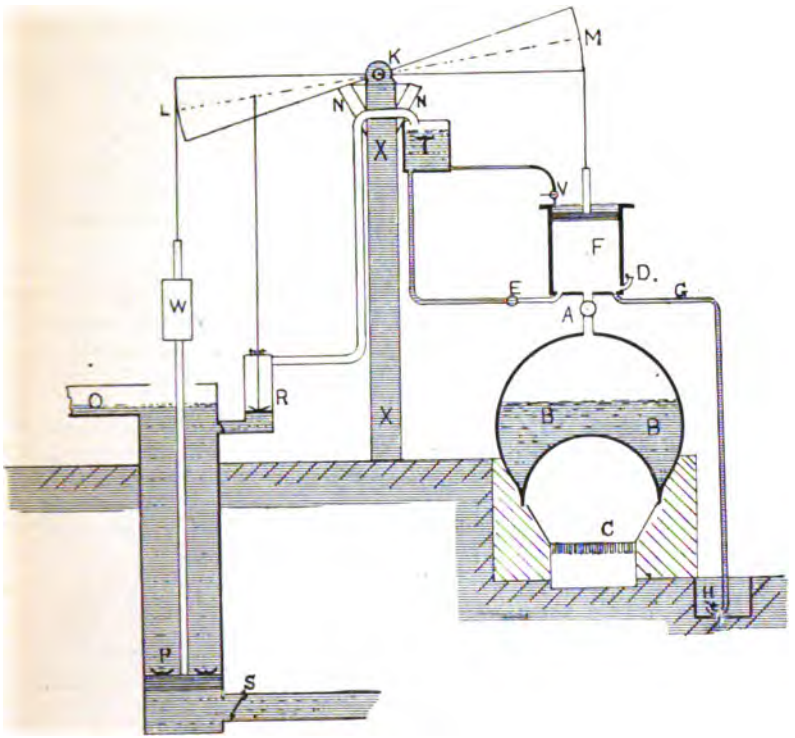


FIG. 4.—Newcomen's Pumping Engine.

has moved through about two-thirds of the upward stroke, the cock *A* is closed, and the steam inside the cylinder expands to fill the greater volume given it by further movement. Just before the end of the upward stroke, the injection cock *E* is opened, admitting cold water into "the heart of the steam," which is almost instantly condensed to water. The mixed condensed steam and water falls to the bottom of the cylinder, leaving the upper space, below the piston, practically clear of pressure of vapour (in other words, an almost complete *vacuum* is formed). The atmospheric pressure,

or weight of the atmosphere, presses on top of the piston and forces it downward, due to a difference of pressure of about 11 lb. (14·7–3·7) per square inch of piston area. This force, pulling the end *M* of the beam downwards, raises the weight and the pump bucket with the water above it, and discharges some water through the opening at *O*. Just before the end of the downward stroke, the steam is again admitted to the cylinder, and, cushioning the piston, brings it gently, and without shock, to rest at the end of the stroke; the water is expelled from the cylinder into a hot-well, or tank, *H*, through a pipe *G* and non-return valve.

The smaller pump *R* simply serves to keep the tank *T* supplied with water for injection purposes, and for sealing the piston to prevent air-leakage past its edge into the cylinder. An escape valve, *D*, weighted to about 2 lb. per square inch above the pressure of the atmosphere, prevents any undue pressure accumulating in the cylinder, and also lets out any excess of the injection water.

The workmanship of this engine was necessarily extremely rude; the engine derived its power principally from the atmospheric pressure on top of the piston, and was *single-acting*—that is, the power was obtained on the down-stroke only, and the preponderance of weight at *L* produced the upward movement. It was soon found that if too much injection water was used, although a much lower pressure was obtained by more rapid condensation, a large quantity of steam was wasted in reheating the cylinder at each upward stroke; the fuel expenditure increased when the temperature of the outgoing eduction water was reduced below about 150° F. The range in temperature of the steam in the cylinder was about 212°-150°, or 62° F., which corresponds very closely with that adopted for economical working in the H.P. and M.P. cylinders of to-day. Some of these engines had steam cylinders, made of brass, 6 feet in diameter and 9 feet piston stroke.

About 1765 James Watt began to make improvements in the steam engine by introducing a separate vessel, called a *condenser*, in which the steam could be condensed (*H* in Fig. 5). The object of this is to keep the cylinder warm, so that the steam on first entering from the boiler should not be so freely used and condensed in heating the cylinder to its own temperature, and therefore be able to act more promptly in raising the piston. At the same time, it is necessary to keep the condenser continually pumped clear of the condensed steam and water used for cooling it. The cooling water weighs roughly 40 to 60 times the weight of the steam which it is used to condense; and, in addition, a large volume of air and other uncondensable gases enters the cylinder and condenser with the steam and water, and as this volume is very large compared with the water, the pump used for the purpose

of clearing the condenser is called the *air pump*, and is shown to the left of the condenser at *N*.

In a further endeavour to keep the cylinder warm, and to prevent cold air from entering it above the piston as in Newcomen's engine, Watt encased the cylinder in another and outer cylinder, as shown in the figure. In so doing he closed the top of the cylinder, except where the piston rod *R* passes through it. At this place a stuffing box and gland is fitted around the rod, so

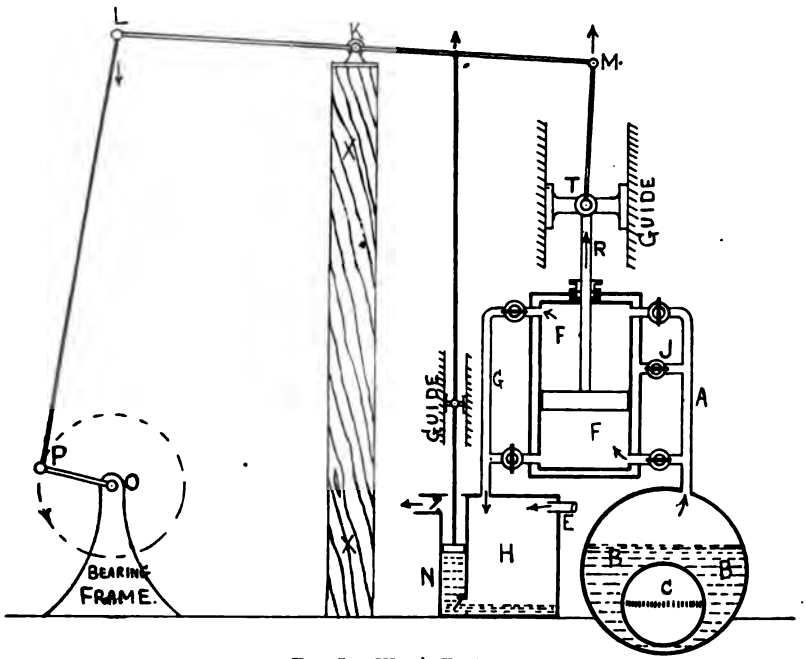


FIG. 5.—Watt's Engine.

that neither can steam escape from the cylinder nor can air leak into it. The engine now became *double-acting*, and independent of the atmospheric pressure, for, by an arrangement of taps or valves, steam could be admitted above the piston to force it downwards as well as upwards. This arrangement of taps is shown in the figure, with each one opened or closed as necessary for the upward stroke of the piston; steam then enters through *A* from the boiler, below the piston, and the escaping or exhaust steam above the piston passes away through the pipe *G* to the condenser *H*.

With this arrangement the *steam jacket* was also used with the same object, that of keeping the cylinder warm, by allowing steam to enter by a pipe and valve, *J*, into the *jacket space* between the inner and outer cylinders. (In some cases it was arranged for the steam to pass through the jacket on its way to the cylinder; this was a common but wasteful method, and is now seldom used.) The water condensed from the jacket steam, in its endeavour to keep the cylinder warm, is taken away by a drain pipe (not shown) to the condenser or open air.

The top end of the piston rod was connected with the end of the see-saw beam *LKM* by a *connecting rod TM*, with pin joints at each end. To keep the piston rod in line with the centre of the piston and cylinder, and to prevent its being bent by oblique forces, a *crosshead*, *T*, was also fitted and guided by two parallel bars, called the guides, or *guide*.

About 1783 Watt applied his engine to rotate a shaft, *O*, at first through a sun-and-planet motion, but ultimately through a connecting rod, *LP*, and a crank, *OP*. An engine of this type was used by John Fitch of Connecticut, about 1786, for moving a skiff by paddles.

With only one cylinder, and when the piston is on the top or bottom of its stroke, called the dead points or *dead centres*, it is evident that the engine can exert no force to rotate the crank, and therefore a *flywheel* was fitted on the shaft. The heavy flywheel, gathering energy of rotation during the working parts of the strokes, gives out some of its energy at the dead centres by continuing its motion, and thus tends to promote a uniform speed of rotation of the shaft. (The flywheel is seldom fitted to marine engines, but it is a common feature of petrol and other small engines.)

Later an improvement was made which, to a large extent, rendered the flywheel unnecessary. By making two cylinders to each engine, and coupling their pistons respectively with two cranks side by side and at right angles to each other on the same shaft, one piston is always in a position to produce motion. This arrangement is very convenient for starting and reversing heavy engines.

In all the very early engines, the admission of steam to, and the opening of the escape or exhaust from, the cylinders were controlled by hand. Later the valves were worked by the engine

itself, so that after the engine was once started it continued working automatically, and was, therefore, self-contained. Still later, the slide valve was introduced, by which both admission and exhaust are controlled by a single valve. (The slide valve is fully explained later, in its proper place.)

In 1801 Symington constructed the first *direct-acting engine*, in the form which is now most commonly used, thus dispensing with the beam, as shown in Fig. 6. The piston rod is here shown connected directly with the crank through the medium of the connecting rod. By turning the picture into various positions, we get—a *horizontal engine*, when the piston moves to and fro in a horizontal line; a *vertical engine*, when the crank is at the top and the cylinder at the bottom; a *vertical inverted engine*, commonly called the vertical engine, and generally used for marine purposes, when the cylinder is at the top and the crank at the bottom; or an *inclined engine*, when the piston works to and fro in some line other than vertical or horizontal.

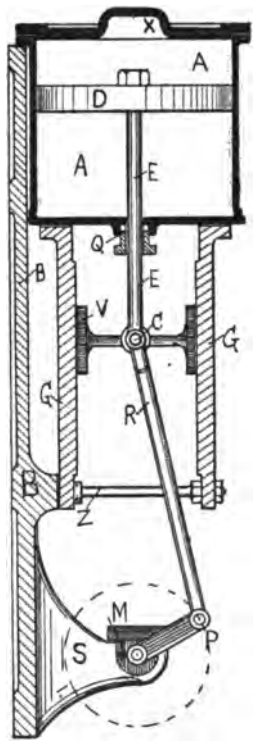


FIG. 6.—Direct-acting Engine.

A, Cylinder.
B, Frame.
C, Crosshead.
D, Piston.
E, Piston rod.
G, Guide.
M, Main bearing (for shaft).
O, Centre of shaft.
P, Crank.
Q, Packing gland.
R, Connecting rod.
S, Part of frame *B* (and holding *M*).
V, Slipper (part of *C*).
X, Cylinder cover.
Z, Stay between guides.

Historical Notes. — About 1781 Hornblower suggested taking the exhaust steam from an engine exhausting to the atmosphere and utilising it in another cylinder. This system of *compound*, or stage-expansion, *engines* did not come into general use until about 1860. In 1782 Watt improved the movement of the admission and exhaust valves so as to cut off the supply of steam to the cylinder before the piston finished its stroke, thus using the steam expansively. *Expansive working* was a feature of all the early engines, but was not entirely understood. Owing to

the difficulty of reversing, the early locomotives were nearly all practically non-expansive working, and it was not until about 1840 that *lap* was added to the *slide valve*, and an increased economy of about 30 per cent was obtained by the consequent increased work done during expansion.

In 1787-88 Symington and Taylor built a twin boat propelled by *paddle wheels* worked by a steam engine of about one horse-power, which obtained a speed of five miles an hour. From this early effort grew the extensive use of paddle-wheel propelled ships, and it is noteworthy that a steam-paddle frigate took part in the bombardment of Acre in 1840.

In 1785 the *screw propeller* was proposed by Watt, and patented by Bramah for the propulsion of vessels, but was first practically used by Stevens (U.S.A.) in a small boat driven by a double-acting engine. Stevens also invented *closed ashpit draught*, a system which is now becoming extensively used in the Navy, although very common in the mercantile marine for some thirty years past. It was only, however, in 1836 that serious efforts were made, by Petit Smith, with screws. He succeeded in attaining about 9¼ knots[1] with the *Archimedes*. This favourable result led to the introduction of screw propellers in H.M.S. *Mermaid* and *Rattler*.

The early marine engines worked at a pressure, by gauge, of about 4 or 5 lb. per square inch; and in 1860 the pressure had risen to about 20 or 30 lb. per square inch. *Jet condensers* were then usual; but the lime scale formed in the boilers, and left on the heating surfaces, from the use of sea water, prevented an increase of pressure with safety. The introduction of the *surface condenser* about 1860, by which the boilers received fresh or nearly fresh feed water, allowed an increase of pressure to about 35 lb. Next an improvement in the shape of boiler, from the square box to the cylindrical type, allowed a further increase to 60 lb., and compound (two-stage expansion) engines also came into fashion. About 1887 the introduction of the *evaporator* for making sea water into fresh water, the general improvement in manufacture of the boilers and machinery, the use of *mild steel* with a higher allowable working stress, and the *corrugated furnace*, allowed a further increase in pressure to about 155 lb., and three-stage (and later four-stage) expansion engines became usual. In 1893 the introduction of

[1] 1 nautical mile = 6080 feet. 1 knot = 1 nautical mile, or sea mile, per hour. Thus 9¼ sea miles per hour = 10·65 land miles per hour = 9¼ knots.

water-tube boilers allowed a further increase of pressure, which gradually rose to about 300 lb. in 1897. Beyond this no further increase has been made for boilers, but the pressure for engines has risen to about 270 lb.

Types of engines now merely of historical interest, but still to be found in old vessels, are the oscillating engine, trunk engine (a form of which is now commonly used for oil-gas engines), and the return connecting-rod engine. The horizontal direct-acting engine generally took the place of the above for naval purposes, but was soon superseded by the vertical inverted engine, which in its turn is giving way to turbines.

Stage Expansion.—If there are two or more cylinders to one engine (which is commonly the case) and the steam passes into each

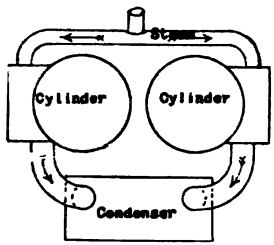


FIG. 7.—Single-stage Expansion Two-cylinder Engine.

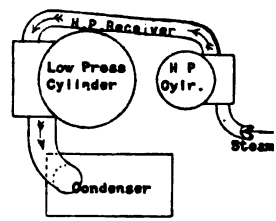


FIG. 8.—Two-stage Expansion Two-cylinder Engine.

cylinder *directly* from the boiler, and is exhausted from each cylinder *directly* into the atmosphere or condenser, the engine is a *simple* or *single-stage expansion* engine. A diagram of the course of the steam for two cylinders is shown in Fig. 7.

If the steam passes into one cylinder from the boiler, and then passes into another cylinder, as shown in Fig. 8—the steam is evidently expanded in two stages and does work successively in two cylinders,—the engine is called a *two-stage expansion* or *compound engine*; but this applies to all engines with two or more stages of expansion. (The vertical inverted cylinder compound engine was introduced by John Elder about 1854.)

If the steam passes through three cylinders in succession, the engine is called a *three-stage* or triple expansion engine. Similarly for four cylinders in succession, the engine is called a *four-stage* or quadruple expansion engine. And so on, for five or more cylinders.

The course of the steam through a three-stage expansion engine

with three cylinders is shown in Fig. 9. The *receivers* are simply pipes connecting up successive cylinders; they receive the exhaust steam from a higher stage, and pass it on to the next lower stage, until finally the exhaust steam enters the condenser through the eduction pipe.

Cycle of Operations.—The three primary elements of a marine steam plant are the boiler, the engine or mechanism, and the condenser. The cycle of operations is started by raising steam from water in the *boiler*; this steam is conveyed away, through the boiler stop valve and the *steam pipe*, to the engine-regulating valve and the

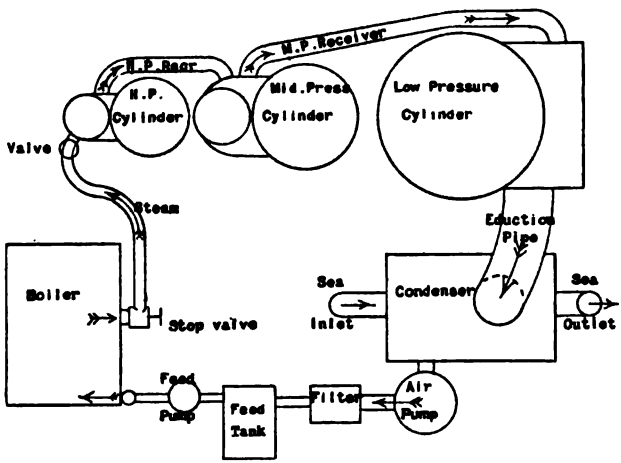


FIG. 9.—Three-stage Expansion Engine and Cycle of Operations.

engine; in the engine the steam does work, by virtue of its heat energy, and is exhausted to the condenser; in the condenser the steam not already condensed during its passage through the engine is condensed to water by the cooling action of the injection water, or of the *circulating water* in a *surface condenser*; from the condenser the condensed water is pumped by the *air pump* into the *feed tank*, and the air, except a moderate quantity which remains in solution in the water, is allowed to escape; finally, the *feed pump* takes the water from the feed tank and delivers it into the boiler, where another similar cycle of operations begins.

In some cases the water is heated, before being returned to the boiler, by a *feed heater* or an economiser (see pp. 110 and 135). Some portion of the water is lost by leakage during the cycle of

operations, and to "make up" this loss an *evaporator* is used to convert sea water into fresh-water vapour or steam; this vapour is conducted to the condenser, and is condensed with the engine exhaust steam.

The above summary gives all the principal parts of an ordinary and simple steam plant; but there are also a number of other useful and essential fittings which are described in detail in other chapters, with the elements to which they belong.

CHAPTER II

UNITS—FORCE, WORK, POWER, ENERGY, PRESSURE, ETC.

In all machines, of which the steam engine is one, a force is exerted at one part which overcomes a resistance at another part; motion is produced and work is done. For the measurements of these forces and resistances some definite units are necessary for comparing the relative values in each machine, and also for comparing one machine with another. These units are given below, and cover all the ordinary requirements of marine and mechanical engineering. A short explanation is also given where it is considered necessary.

Length (L, *l*).—The standard of length is *one yard*, but in engineering the unit is one-third of a yard, or *one foot*.

Time (*t*).—The unit of time is *one second*.

Velocity (V or *v*).—The unit of velocity is *one foot per second*.

The motion of a body is *uniform* when it passes through equal spaces in equal times; it is *variable* when it passes through unequal spaces in equal times, or through equal spaces in unequal times. If the motion is an increasing one, it is *accelerated*; but if it is a decreasing one, it is generally termed *retarded* or *decelerated*.

For the proper measurement of velocity, the *time interval* must be very small. Thus a train may cover the distance between equally spaced stations in equal times, and therefore with an equal *average* velocity from station to station; but the velocity is not uniform, because the actual velocity of the train, from instant to instant, varies considerably.

Volume (V or *v*).—A bounded part of matter is called a body; and the part of space which a body occupies is called its *volume*.

The British unit of capacity, or volume, is *one gallon*, which

contains exactly 10 lb. weight, at sea-level at Greenwich, of *pure water* at the standard temperature of 40° F.

Mass (M or m).—The quantity of matter contained in a body is called its *mass*. The unit of mass is a constant quantity, and does not change with its position relatively to the earth's surface.

Weight (W or w).—The unit of weight is *one pound*, which is equal to the weight of a standard mass, or quantity of matter, *in vacuo*.

The weight of a body, although it may be considered constant for engineering purposes, changes with its distance from the centre of the earth. The acceleration of velocity due to gravity (g) is found by careful experiment to vary, in exactly the same proportion as the weight, when both are measured at the same positions relative to the centre of the earth. The mass of a body is always equal to its weight divided by the gravitational acceleration ; or—

$$m = \frac{w}{g}.$$

At sea-level at Greenwich, the acceleration due to gravity is approximately equal to 32·16 feet per second in each second, and therefore, for ordinary engineering purposes—

$$m = \frac{\text{weight of the same body}}{32{\cdot}2}, \text{ approximately.}$$

Force.—A body which is at rest (*i.e.* remaining at the same place in space) cannot change its position without a cause; nor can a moving body change the conditions of its motion without a cause. Such a cause is called *force*.

The engineering unit of force is *one pound*, and is the force that can balance the weight of 1 lb. at sea-level at Greenwich.

An *effort* is said to be applied when a force is caused to act tending to produce motion of a body. A *resistance* is a force which acts opposite to the direction of, and is called into play by, an effort.

Force also changes the "state of rest" of a body; that is, it may call an increased frictional resistance into play without either reaching the limiting friction or just producing motion.

If a force of 1 lb. produces motion of a body, it produces a certain rate of change, or acceleration, in its velocity. The unit of force may also be defined as that force which can give unit acceleration to a body of unit mass. It is evident that this unit of force is a constant quantity; because the unit of acceleration, being 1 foot per second in each second, does not vary; and the unit of mass, being also a constant quantity, does not vary. This definition of force is in agreement with the one previously given, and is merely another way of defining the same quantity.

Density.—The density of a substance is the mass contained in one unit of volume of the substance. The density varies with the temperature and the pressure to which a substance is subjected.

All substances are compressible; but the practicable compression of many solid and liquid substances is so small that the change in density may usually be neglected.

Specific Density, or Gravity.—The relative density, or specific gravity, of a substance is the ratio of the density of a substance to that of *pure water* at a standard temperature. This temperature is often taken at 39° Fahr., which is the temperature of water at about its maximum density. The relative density of a gas is usually taken as the ratio of the density of the gas to that of *air* at 32° Fahr., and at a pressure of one atmosphere (14·7 lb. per square inch). (*Note.*—In chemistry and physical experiments and data, *hydrogen* instead of *air* is used for the standard of comparison.)

Specific Volume.—The specific volume, or bulkiness, of a substance is the number of units of volume occupied by unit mass of the substance.

For steam, it is measured by the number of cubic feet to 1 lb. by weight at any given pressure.

Relative Volume.—This is the ratio of the volume of the steam produced to that of the water from which it is generated.

One pint of water, weighing 1¼ lb., when completely evaporated into steam, fills 1643 pints under normal atmospheric pressure. If this steam be compressed under a pressure of about 21 atmospheres, the *relative volume* becomes about 90 pints. If the pressure be decreased, so as to become about two-fifteenths the normal atmospheric pressure (corresponding to a vacuum of 26 inches on the mercury barometer), this steam then fills about 11,000 pints, and its relative volume is about 11,000. (See column 3 of table, p. 18.)

Work.—Work is said to be done when motion is produced against resistance. The product of the resistance in terms of units of force, and the distance in terms of units of length, through which the resistance is overcome, is therefore a measure of the work done against the resistance.

The unit of work is *one foot-pound* (1 foot-lb.).

Other units are sometimes used in engineering which are self-explanatory, such as foot-tons, inch-tons, etc.

Power.—The power of an engine is the rate at which it can do work, *i.e.* the quantity of work it can do in a certain time.

The unit of power is *one foot-pound-second*; but as this unit is

small, it is more convenient to use a derived unit of power—this is *one horse-power* (H.P.), which is equal to 33,000 *foot-lb.* of work per *minute.*

One *cheval vapeur* (French H.P.) is equal to 32,500 foot-lb. per minute.

Energy.—The capacity of bodies for doing work is called energy, and a body is said to contain more energy the more work it can do, regardless of time. The terms *energy* and *power* must not be considered to mean the same; power has relation to time. Thus 1 ton of coal may, by being burnt in a sufficiently large boiler, make sufficient steam to produce 1200 horse-power for 1 hour; whereas, when burnt in a small boiler, it may produce 120 horse-power for 10 hours—the energy contained in the coal being the same in both cases.

Energy appears in various forms—mechanical energy, heat, electrical energy, etc.; but they are all mutually convertible, as we shall see later.

Pressure.—When a force is acting it is generally distributed over an area (even when the force is acting on a point or line, because both a point and line may be considered to have some area); the unit of area in engineering is taken either as 1 square foot or as 1 *square inch.* For pistons, plungers, cylinders, etc., used in fluid machines, the unit is commonly taken as 1 square inch.

The part of the force exerted on each square inch is called the pressure (p), or pressure per square inch (or the intensity of pressure per square inch, or unital pressure), and it is evident that, if the pressure is uniform, the pressure on each square inch being equal, this pressure p, multiplied by the area A in square inches, is equal to the force exerted, F; or—

$$p \times A = F.$$

Atmospheric Pressure.—The pressure of the atmosphere depends on the relative density of the atmosphere, and at Greenwich, corrected to sea-level, is 14·7 lb. per square inch when the mercury barometer stands at 30 inches. In other words, when the atmospheric pressure is 14·7 lb. per square inch, it will support a column of mercury 30 inches in height.

The pressure of the atmosphere is actually equal to the weight of a vertical column of the atmosphere, whose height is equal to the height of the atmosphere at the place of measurement and

whose horizontal area is the unit of area on which the pressure is exerted.

Absolute Pressure.—Absolute pressure (p) is measured from the *zero pressure* of the atmosphere; it is therefore = (gauge pressure + 14·7) lb. per square inch.

The nought (0), or zero, shown on pressure, combined pressure and vacuum, and vacuum gauges, always corresponds to atmospheric pressure (14·7 lb. per square inch). Thus if a boiler pressure-gauge shows 60, it means 60 lb. *above* atmospheric, or 74·7 lb. *absolute* pressure. When the vacuum gauge shows 30 inches, it means there is zero absolute pressure, or 14·7 lb. *below* atmospheric. For purposes of general application, an *increase* of vacuum of 2 inches represents a *decreased* pressure, below atmospheric, of 1 lb. per square inch.

Mechanical Energy.—There are two kinds of mechanical energy. One is *potential*, or that of position, and is best illustrated by that of a weight suspended at some point above the earth. The other is *kinetic*, and is representative of any body in motion, and may be illustrated by a rotating flywheel. Both the weight and the flywheel possess capacity to do work, or energy: the former in falling to some lower level, and the latter by reason of its motion.

When an engine is rotating a shaft or performing other work, with a uniform rate of motion and overcoming a uniform resistance, then—

Energy developed = work done (in the same units of measurement);

but if the motion be accelerated while the resistance remains uniform, then—

$$\text{Energy developed} = \text{work done} + \left\{\begin{array}{l}\text{kinetic energy accumulated in}\\ \text{the mechanism.}\end{array}\right.$$

The acceleration may be either positive or negative, the velocity being respectively either increased or decreased. If the acceleration be negative, then work is done at the expense of the kinetic energy previously accumulated in the mechanism, which is correspondingly reduced, and—

$$\text{Energy developed} = \text{work done} - \left\{\begin{array}{l}\text{kinetic energy taken from the}\\ \text{mechanism.}\end{array}\right.$$

As an illustration, suppose a weight, W, is lifted through a height, H, then the work done = W × H in foot-lb. The weight

then possesses potential energy of W × H foot-lb. due to its position. As the weight falls it loses potential energy and accumulates energy of motion, or kinetic energy. In falling through any distance H, the velocity V accumulated by the falling body is found by many concise experiments to be (where g = acceleration due to gravity = 32·2 feet per second per second)—

$$V = \sqrt{2g} \cdot \overline{H},$$

or

$$V^2 = 2g \cdot H,$$

and therefore

$$H = \frac{V^2}{2g}.$$

Multiply both sides of this equation by W; then—

$$W \cdot H = \frac{W \cdot V^2}{2g}.$$

Potential energy = $W \cdot H = \frac{1}{2}m \cdot V^2$ = kinetic energy.

If the weight falls through only a part, h, of its height, H, then at this point we obtain the equation (where v is the velocity accumulated at the end of the fall h)—

$$W \cdot H = W(H - h) + W \cdot \frac{v^2}{2g}.$$

Or,

$$\left.\begin{matrix}\text{Potential energy} \\ \text{available}\end{matrix}\right\} = \left\{\begin{matrix}\text{potential energy} \\ \text{not utilised}\end{matrix}\right\} + \left\{\begin{matrix}\text{kinetic energy} \\ \text{accumulated.}\end{matrix}\right.$$

Properties of Saturated Steam.

p	t	$\frac{v}{s}$	w	v	S	L	L+S=H	p_0
Gauge Pressure.	Boiling-point Fahr.	Relative Volume.	Weight of 1 cub. ft.	Volume of 1 lb. Weight.	Sensible Heat.	Latent Heat.	Total Heat of Evaporation of 1 lb. Steam from 32° Fahr.	Absolute Pressure.
	°		Lb.	Cub. ft.	B. Th. units.	B. Th. units.	B. Th. units.	Lb. per sq. in.
Inches of mercury. 28·56	90·4	28740·0	0·00217	462·2	58·4	1051·1	1109·5	0·70
27·95	102·1	20582·0	0·00303	331·8	68·9	1044·7	1113·1	1·0
25·9	126·3	10721·0	0·00584	172·9	93·2	1027·3	1120·5	2·0
23·87	141·6	7322·0	0·00852	118·1	108·8	1016·4	1125·2	3·0
21·83	153·1	5583·0	0·01117	90·0	120·6	1008·1	1128·7	4·0
19·79	162·3	4527·0	0·01378	73·0	130·0	1001·5	1131·5	5·0
17·75	170·2	3813·0	0·01636	61·5	138·1	995·8	1133·9	6·0
15·71	176·9	3298·0	0·01891	53·2	144·9	991·0	1135·9	7·0
13·67	182·9	2909·0	0·02144	46·9	151·0	986·7	1137·7	8·0
11·63	188·3	2604·0	0·02394	42·0	156·6	982·8	1139·4	9·0
9·57	193·3	2358·0	0·02644	38·0	162·7	978·2	1140·9	10·0
7·55	197·8	2157·0	0·02891	34·8	166·3	976·0	1142·3	11·0
5·51	202·0	1986·0	0·03138	32·0	170·6	973·0	1143·6	12·0
3·47	205·9	1842·0	0·03383	29·7	174·5	970·2	1144·7	13·0
1·43	209·6	1720·0	0·03626	27·7	178·4	967·5	1145·9	14·0
0	212·0	1643·0	0·03793	26·5	180·2	965·8	1146·5	14·7 atmosphere
Pounds per square inch. 0·3	213·1	1610·0	0·03869	26·0	180·5	965·0	1146·9	15·0
2·0	218·5	1457·0	0·04279	23·1	186·5	961·4	1147·9	16·7
5·0	227·1	1248·0	0·05000	19·8	195·0	955·4	1151·1	19·7
7·0	232·2	1139·0	0·05474	18·2	200·3	951·8	1152·1	21·7
10·0	239·4	1009·0	0·06184	16·1	208·3	946·7	1155·0	24·7
15·0	249·7	818·2	0·07355	13·5	217·7	939·9	1158·1	29·7
20·0	258·9	732·8	0·08513	11·7	227·8	933·1	1160·9	34·7
30·0	273·9	577·6	0·1080	9·2	243·2	922·3	1165·5	44·7
40·0	286·5	477·7	0·1305	7·6	256·2	913·2	1169·4	54·7
50·0	297·5	408·0	0·1529	6·5	267·3	905·4	1172·7	64·7
60·0	307·2	356·4	0·1750	5·7	277·1	898·5	1175·6	74·7
70·0	315·8	316·7	0·1969	5·1	286·0	892·3	1178·3	84·7
80·0	322·8	285·2	0·2187	4·6	294·2	866·5	1180·7	94·7
90·0	331·0	259·8	0·2404	4·2	302·9	880·3	1182·9	104·7
100·0	337·7	238·2	0·2609	3·8	308·5	876·5	1184·0	114·7
110·0	343·9	220·1	0·2833	3·6	314·8	872·0	1186·8	124·7
120·0	349·8	204·8	0·3046	3·3	320·8	867·8	1188·6	134·7
140·0	360·6	179·9	0·3469	2·9	332·0	859·9	1191·9	154·7
150·0	365·6	169·5	0·3680	2·7	337·2	853·6	1193·5	164·7
160·0	370·4	160·4	0·3889	2·5	342·1	852·8	1194·9	174·7
180·0	379·3	144·8	0·4307	2·3	351·3	836·3	1197·6	194·7
200·0	387·5	132·1	0·4721	2·1	359·8	840·3	1200·0	214·7
210·0	391·5	127·5	0·4912	2·0	365·2	836·1	1201·3	224·7
220·0	395·2	122·9	0·5117	1·9	369·3	833·2	1202·5	234·7
240·0	402·5	113·0	0·5528	1·8	376·7	827·9	1204·6	254·7
260·0	409·2	106·2	0·5936	1·7	383·3	823·0	1206·7	274·7
280·0	415·6	99·4	0·6344	1·6	390·5	818·2	1208·7	294·7
300·0	421·7	93·5	0·6752	1·4	396·9	813·6	1210·5	317·7
320·0	427·5	88·6	0·7157	1·35	403·1	809·2	1212·3	334·7
350·0	435·8	81·3	0·7766	1·3	411·9	802·9	1214·8	364·7
385·0	444·9	73·1	0·8144	1·17	419·0	798·6	1217·6	399·7
485·3	466·6	55·8	1·065	0·94	444·3	779·9	1224·2	500·0
p	t	$\frac{v}{s}$	w	v	S	L	H=L+S	p_0

CHAPTER III

NATURE AND EFFECTS OF HEAT

The Steam Engine is a Heat Engine.—The steam engine is not only a machine for the transmission of force, but it is also a *prime mover* which receives its energy from a natural source (viz. the combustion of fuel), and is a heat engine. The steam, derived from the combustion of the fuel, is merely a medium for conveying the heat energy derived from the fuel to the engine, where it is converted into mechanical work. For the proper understanding of the steam engine it is necessary to know something of the nature and properties of heat, and the effects produced on water by the addition of heat.

Nature of Heat.—Only one definite statement can be made of the nature of heat: it is *a form of energy*, which can be conveyed under certain well-defined conditions from one body to another body or substance.

Heat is now generally supposed to consist in the rapid vibratory motion of the particles or molecules of which a substance is composed, and in which the more rapid the molecular motion becomes the greater is the energy in the substance. From this it may be considered that heat is a *condition of matter*, and a condition which can be transferred to another substance. The properties of heat are known to us from the effects produced by the transference or flow of heat from one body to another, and, in accordance with natural laws, a flow of heat can only take place from a hotter to a colder body.

Temperature (T or *t*).—The temperature is a measure of the condition or *quality* of one part of a body in regard to its being more or less hot than another part of the same body or of another body.

Temperature must not be confounded with heat, which is a *quantity*. Thus two bodies of water may be of the same temperature, but the larger body contains the greater quantity of heat, due to its greater weight. One substance

does not directly impart its temperature to another substance; but it can transfer some of its heat, and in this way raise the temperature of the other substance until a common temperature is obtained, which is always lower than the originally higher temperature.

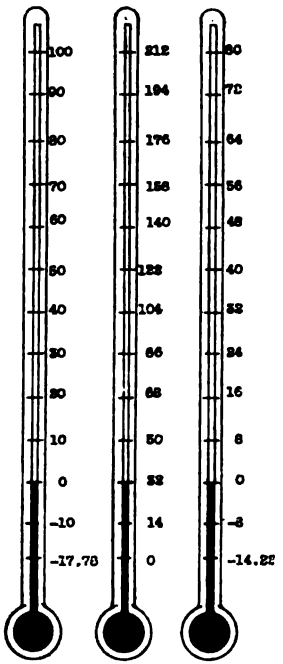


FIG. 10.—Thermometers.

Thermometers. — The instrument used for measuring temperatures is called a thermometer, and the thermometer generally used in engineering, in English-speaking countries, is graduated on the Fahrenheit scale.

The Centigrade, or Celsius, scale is used in countries where the metrical system is adopted, and generally for physical and chemical experiments. The Réaumur system is used in Russia and in some remote parts of Germany; but in both these countries there is a general disposition to adopt the Centigrade system, particularly in Germany.

In each thermometer the boiling and freezing points of pure water, under the normal atmospheric pressure of 14·7 lb. per square inch, are first determined, and the distances between these points respectively for each thermometer are divided up into degrees. The number of degrees correspond to 180, 100, and 80 for Fahrenheit, Centigrade, and Réaumur respectively. The freezing-points are zero for both Centigrade and Réaumur, and 32 degrees above zero for Fahrenheit. Water, therefore, boils (under normal atmospheric pressure) at 212°, 100°, and 80° respectively on the Fahrenheit, Centigrade, and Réaumur thermometers. A diagram of the three systems is shown in Fig. 10.

To convert R. into C. : $R. = \frac{4}{5}C.$, and $C. = \frac{5}{4}R.$
" C. into F. : $F. = \frac{9}{5}C. + 32$, and $C. = \frac{5}{9}(F. - 32)$
" F. into R. : $F. = \frac{9}{4}R. + 32$, and $R. = \frac{4}{9}(F. - 32)$

Effects of Heat.—When heat passes from one substance to another, three distinct changes are effected in one, and at the same time similar changes in an opposite sense take place in the other. The heat energy transferred to the substance causes changes in the substance itself. Part is absorbed in (1) doing *internal work*, by increasing the vibratory motion of the molecules and tending to increase its volume. This increased vibratory motion is accompanied by (2) an increase in temperature. Another part (3) is absorbed in

doing work on external bodies, generally termed *external work*, but only when an increase in volume is allowed to take place, and thus the resistance or pressure of other surrounding bodies is overcome through a space equal to the increase in volume. If no change in volume takes place (and this can only happen in the case of comparatively easily compressible substances such as gases and certain liquids, but unusual in metals), and the same quantity of heat passes as before, then a further rise in temperature takes place, and heat is added to the *internal energy* of the substance exactly corresponding to the work done in resisting its expansion.

All substances, under certain conditions of pressure and temperature, can be made to take either the solid, the liquid, or the gaseous state, and therefore, by adding heat to or taking heat away from any substance, either one of the following three effects may be produced: (1) expansion or contraction; (2) liquefaction or solidification; (3) evaporation or liquefaction.

Expansion of Solids and Liquids.—When heat is added to a substance, solid or liquid or gaseous, it expands to a more or less degree corresponding to the difference in temperature. (There are a few exceptions to this rule: india-rubber being, perhaps, the most common in use; water also contracts when raised from freezing temperature to about 39·1° Fahr., but above this temperature expands.) A solid and practically incompressible substance such as a metal expands in three directions—length, breadth, and thickness; but for general purposes of engineering a knowledge of its expansion in one direction is sufficient. It is approximately the same amount per unit of dimension in each of the three directions. The *linear* expansion of solids, or increase in length, is that generally considered and measured.

The *coefficient of linear expansion* is the increase of length per unit of length when the temperature is increased 1°. This coefficient increases with the temperature, but for ordinary purposes may be considered constant in value.

EXAMPLE.—A steel rod, whose length is 1 at 32° Fahr., increases to a length of 1·0011447 at 212° Fahr.: what is the coefficient of expansion?

$$\text{Coefficient} = \frac{0{\cdot}0011447}{180} = 0{\cdot}000006415.$$

Also, what is the increase in length of a steel tube, 7 feet long, when the temperature is increased from 32° to 732° Fahr.?

For each degree, 1 foot becomes $(1 + 0{\cdot}000006415)$ feet
„ 700 degrees, 1 „ „ $(1 + 700 \times 0{\cdot}000006415)$
„ 700 „ 7 feet become $(1 + 700 \times 0{\cdot}000006415) \times 7$
Linear expansion $= 4900 \times 0{\cdot}000006415$ feet
$= 0{\cdot}0314335$ feet
$= 0{\cdot}377$ inch

Liquids expand to a moderate extent, and it is only necessary to measure the increase in volume, which is three times (very approximately) the linear expansion. Thus if x be the coefficient of linear expansion, then the volume of a cube whose side is originally 1 becomes $(1+x)^3 = 1 + 3x + 3x^2 + x^3$. The value of $3x^2 + x^3$ is so small that it may conveniently be neglected, so that the new volume $= 1 + 3x$ very approximately.

In a mercury barometer, the glass container or tube has a very small coefficient of expansion, while the mercury has a very large one; the expansion of the glass can generally be neglected, while the mercury, increasing in *volume*, apparently rises three times the amount due to its linear expansion. The above is a very common case with glass vessels and liquids; but if a metal vessel is used, an actual depression may be sometimes observed in the height of certain liquids when heated, because the coefficient of expansion of the metal vessel is so much greater than (more than three times) that of the liquid.

The difference in the expansion of two metals can be readily observed by riveting the opposite ends of two different metals together, and then heating them; the difference of expansion produces unequal increases in length between the rivets, and causes some curvature, in which the metal with the greater coefficient takes the outside, and therefore longer, curve.

Allowance for expansion (actually for unequal expansion of two bodies) is very common and almost universal in engineering structures. The rails for tram and other railways are generally placed with their ends a certain distance apart, so that when warmed by the sun they can expand lengthways; similarly, steam pipes are fitted with expansion glands or with bends to allow expansion and contraction. Iron bridges, boiler furnaces, and some boiler stays have also some arrangement by which expansion is allowed within certain limits. Smoke-box doors and boiler casings, when made flat, tend to warp when heated, and are now frequently constructed of corrugated plating, so as to maintain their original form and shape.

In the foundry, the cooling of molten metal when run into a mould frequently causes fracture from unequal contraction; for this reason the spokes of a large cast-iron or steel wheel are curved, so that contraction is allowed for when the metal cools. The pattern used for moulding must also be larger than the article to be made. Boiler furnaces and tubes, cylinders and their connections, are subjected to change of form when heated and cooled, and neglect in allowing for the consequent expansion or contraction has frequently resulted in fracture.

Water expands when cooled from 39·1° Fahr. downwards until ice is formed, when further expansion takes place. Ice forming in the mouth of a pipe prevents further expansion of the water within it, and the pipe may burst. For this reason water pipes are generally laid at some depth below the surface of the ground, where they are less subject to cooling effects or changes of temperature and state.

Expansion of Gases.—Unlike solids and liquids, gases expand easily and infinitely, and thus any space or volume to which a gas has access is immediately filled with gaseous vapour; or, when a portion of the gas contained in a chamber is removed by the action of a pump or other means, the gas remaining in the chamber will instantly expand so as to fill the whole volume. Conversely, gas or air can be compressed so as to occupy comparatively very much smaller volume.

In considering the volumetric expansion of air or gas within an engine cylinder, the expression *stroke volume* is commonly used or implied. Referring to a cylinder and piston, such as are shown in Fig. 11, it will be seen that the volume of the gas inside the cylinder and behind the piston = length $AC \times$ area of piston. If the piston be moved to any point, G, the volume = length $AG \times$ area of piston. Thus the volumes of the gas at C and G respectively are represented (for the same cylinder) by the lengths AC and AG. In other words, if the stroke of the piston be represented by any line, AB, it also represents to scale the volume, depending on the area of the piston, which is constant for the same cylinder. Also, if any two points, C and G, be taken in this line, the distance between them represents, to scale, both the stroke of the piston and the volume struck out by it between the two points.

The **First Law of the Permanent Gases**, commonly known as Boyle's or Marriotte's law, states that—

"The volume (V) of a given *mass* of a gas varies inversely as the (absolute) pressure (P) provided the temperature (T_0) be kept constant."

Briefly, this law means that if 2 gallons of a gas are compressed

into 1 gallon, provided no change in temperature takes place, then the absolute pressure will be doubled; and so on. Using the usual notation—

$$V \propto \frac{1}{P} \text{ when } T_0 \text{ is constant,}$$

$$P.V = \text{constant when } T_0 \text{ is constant,}$$

and

$$P \propto \frac{1}{V} \text{ when } T_0 \text{ is constant.}$$

Boyle's law of expansion can be represented graphically. Let us suppose that a given mass of gas fills a certain space behind a piston in a cylinder, as shown in Fig. 11, and take a straight horizontal line, *AB*, to represent the extreme stroke of the piston to some convenient scale, and suppose the piston in the first instance to be at *C*, so that *AC* represents the volume of the given mass of gas. From *A* set up *AD* perpendicular to *AB*, and representing the absolute pressure of the gas in the first instance, termed the *initial absolute pressure*, to some convenient scale; so that 1 inch in height of *AD* represents a certain known pressure per square inch, and so on for each inch and for each part of an inch. Describe a rectangle, *ADEC*; then the area of this rectangle is equal to—

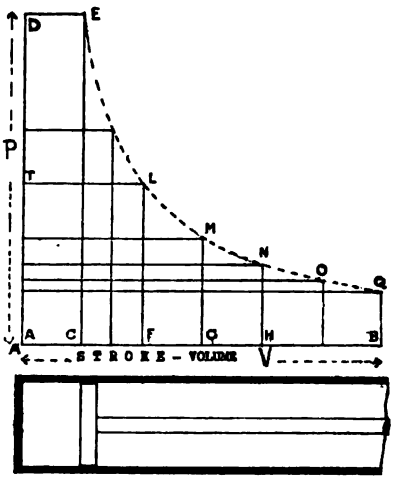


FIG. 11.—Boyle's Law and Stroke-volume.

$$AD \times AC = p \,.\, v.$$

Cut off *CF*, and make it equal to *AC*, so that when the piston is moved to *F* the gas inside the cylinder expands to twice its original volume. Make $AT = \frac{1}{2}AD$. Then the rectangle *ATLF* = rectangle *ADEC* in area, and the line *FL* represents the absolute pressure when the piston is at *F*. Any number of rectangles may be described as shown in the figure, each of which is equal to *ADEC*, and a series of points, *E*, *L*, *M*, *N*, *O*, *Q* . . . may be found such

that their respective vertical heights above AB represent to scale the absolute pressure at those points for the respective volumes AC, AF, AG, AH. . . . By drawing a fair curve through the points E, L, M, N, O, Q . . . a graphical representation is obtained of the absolute pressure in the cylinder for every point in the stroke of the piston.

Thus, $p \cdot v = AD \times AC = BQ \times AB = \mathrm{P} \cdot \mathrm{V} = \text{constant}$.

It must not be forgotten that this law assumes that the expansion is *isothermal* (*i.e.* that no change of temperature takes place as the gas expands). As we shall see later, no work is then done by the expanding gas when the piston moves, unless heat is received into the gas from some outside source.

Gases also expand when heated, or tend to expand, and if the tendency to expand is resisted by the containing vessel or other means, then the pressure is increased because the volume cannot increase. Conversely, if heat is taken away from a gas the pressure decreases, because the gas expands and keeps the volume constant by filling the same space.

Absolute Temperature.—For every decrease of 1° in temperature of air below the freezing-point (32° Fahr.), it loses about $\frac{1}{493}$ part of its volume, and therefore it is assumed that at a temperature of 493° below 32° Fahr., or of 461° below the *zero* of the Fahrenheit thermometer scale, both volume and temperature disappear. This point is taken as the *zero of absolute temperature.* For example, the absolute temperature of the boiling-point of fresh water under ordinary atmospheric pressure is—

$$212° + 461° = 673°.$$

We are now in a position to state the **Second Law of the Permanent Gases**, as formulated by Charles and by Gay-Lussac :—

"Under constant pressure, equal volumes of different gases increase equally for the same increment of temperature. Also, if a gas be heated under constant pressure, equal increments of its volume correspond very nearly to equal intervals of temperature as determined by the scale of a mercury thermometer."

Or $\mathrm{V} \propto \mathrm{T}_0$ when P is constant, and T_0 is the *absolute* temperature.

Thus if 1 gallon of gas is heated until its absolute temperature is doubled, its volume is also doubled, becoming 2 gallons if no change in pressure takes place.

For a given mass of gas, by combining the first and second laws,

$$\mathrm{P} \cdot \mathrm{V} = c \cdot \mathrm{T}_0,$$

where c is a constant depending on the specific density of the gas, and on the units in which P and V are measured.

If P is measured in pounds per *square foot*, V is the volume of 1 lb. (weight) of the gas in cubic feet; and T_0 is the absolute temperature on the Fahrenheit scale.

Then, for air—

$$P.V = 53{\cdot}18\ T_0 \text{ in foot-lb.}$$

EXAMPLE.—At 32° F., P = 14·7 lb. per square inch = 2116·8 lb. per square foot.
V = 12·38 cubic feet, by direct experiment.

Therefore $P.V = 26{,}217$ foot-lb.

From which, when $T_0 = 32°$ Fahr. $= 493°$ absolute temperature Fahr.—

$$c = 53{\cdot}18.$$

It follows from the above that when air, or gas, or steam-gas is heated in a closed cylinder, so that its volume cannot change, an increase in pressure takes place, and the internal (or intrinsic) energy of the gas is increased.

Also that when air, or gas, or steam-gas is heated in a cylinder closed by a movable piston on which a constant pressure is exerted, an increase in volume takes place, and, in addition to the increase in internal (or intrinsic) energy of the gas, external (or visible) work is done by moving the piston against a resistance through a certain distance obtained by the increase in volume.

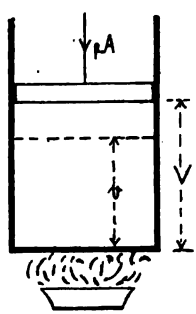


FIG. 12.—External Work during Expansion.

Let v = the original volume,
p = constant pressure per unit of area A,
V = the final volume.

Then (see Fig. 12)—

External work done

= p . A × distance piston is moved
= $p(V - v)$
= pressure × change in volume.

As previously stated, for each increase in temperature of 1° Fahr., air expands $\frac{1}{491}$ part of its volume; and this ratio is found to be approximately true (although not exactly) whether the increase be from 299° to 300°, or from 60° to 61° Fahr., or at any temperature in ordinary use. The coefficient of expansion per unit of volume of ordinary dry air is therefore $= \frac{1}{491} = 0{\cdot}002$ approximately.

Sensible Heat (S).—When a change of temperature alone

takes place in a substance, the quantity of heat used in making the change is called *sensible* heat.

For water, the sensible heat is the quantity required to raise the temperature of 1 lb. weight of water from 32° F., or other temperature, to the point of ebullition, or *boiling-point*; a change of state then begins to take place, and no further increase in temperature occurs (unless the boiling-point is raised by increasing the pressure) until the change of state is completed. Thus a substance may continue to receive heat at certain temperatures, peculiar to the substance and the pressure to which it is subjected, without changing its temperature.

Quantity of Heat.—Various substances have varying capacities for taking in or giving out heat; it requires more or less fuel to raise the temperature of one substance 1° than another. Thus it will cost more fuel to raise the temperature of 1 lb. weight of water through 1° than it will to raise the temperature of the same weight of steel through 1°. It is necessary to have some standard of comparison, and this is the **British Thermal Unit** (B.Th.U.), which is the quantity of heat required to raise 1 lb. weight of pure water at its maximum density (39·1° Fahr.) 1° in temperature. For all practical purposes this quantity may be considered as constant, whether the water be raised from 39° to 40° or from 300° to 301°.

The quantity of heat required to raise unit mass of a substance 1° in temperature is called the **Specific Heat** of the substance.

As we have seen above, the specific heat of water is 1, or unity, and 1 b.t. unit is required to raise 1 lb. of water 1° in temperature. Therefore, in engineering, the specific heat of a substance is the quantity of heat in b.t. units required to raise 1 lb. of the substance through 1° in temperature.

A simple method of determining the specific heat of a substance is to immerse a known weight of it at a known temperature in a known weight of water at a differing temperature. After the two substances have reached the same temperature by communicating heat one to the other, the heat lost in one must be exactly balanced (assuming no external loss takes place) by that gained in the other, and an equation is obtained—

$$\text{Weight} \times \left\{\begin{matrix}\text{rise or fall in}\\ \text{temperature}\end{matrix}\right\} \times \left\{\begin{matrix}\text{spec. heat}\\ \text{of subst.}\end{matrix}\right\} = \text{weight} \times \left\{\begin{matrix}\text{fall or rise in}\\ \text{temperature}\end{matrix}\right\} \times \left\{\begin{matrix}\text{spec. heat of}\\ \text{water (1).}\end{matrix}\right.$$

For accurate measurement and exact determination of the specific heat, various corrections must be made to compensate losses, such as heating or cooling the containing vessel and instruments used (thermometers, etc.).

Changes of State.—The quantity of heat required to change

1 lb. of a substance from a given state into another state without altering its temperature, is called the *Latent Heat* (L) of the substance. For each substance there are two latent heats, corresponding to the two changes of state respectively from the solid to the liquid (or latent heat of fusion) and from the liquid to the gaseous (or latent heat of evaporation); and conversely.

Suppose it is required to change 1 lb. of ice (specific heat of ice = 0·5 approximately) at temperature 0° Fahr. into steam at 212° Fahr., under normal atmospheric pressure; then—

			B.t. units.
Change of temperature,	sensible heat, at 0·5 per 1°,	from 0° to 32° F. =	16
Change of state,	latent heat (of ice),	at 32° F. =	144
Change of temperature,	sensible heat,	from 32° to 212° F. =	180
Change of state,	latent heat (of water),	at 212° F. =	966
		Total quantity of heat required =	1306

It should be noticed that a change of state and a change of temperature do not take place at the same time under the generally accepted engineering conditions, but some changes of state take place at all temperatures in nature; these changes are, however, extremely slow, and do not enter into the present discussion.

Regnault gives 142 b.t. units as the latent heat of fusion of ice. Later experiments by Bunsen and others give the higher value of 144, which is that now generally used by refrigerating engineers. The American Society of Mechanical Engineers recommends a standard unit of ice-melting capacity, equivalent to 288,000 b.t. units per ton (equal to 2000 lb. British weight).

The value 966 is the approximate result of experiments by Regnault.

Evaporation, Ebullition, Boiling-Point.—*Evaporation* is said to take place when vapour is slowly formed at the surface of a liquid, and is discharged into a space not already saturated with such vapour. The rate of evaporation is principally influenced by (*a*) the difference in temperature of the liquid and its surroundings; (*b*) the quantity or degree of saturation or density of similar vapour in the atmosphere or other substance in contact with the surface of the liquid; (*c*) the superincumbent pressure in the space due to other vapours; and (*d*) the area of the surface at which evaporation can take place.

Ebullition, or boiling, takes place when elastic bubbles of vapour are formed in the liquid itself, and are discharged into a space already saturated with similar vapour. The kettle "sings" when the heating releases the air, or other gases, previously in solution in the water. The escape of these bubbles assists materially in the

formation of steam, and, heat continuing to be added, ebullition, or boiling, begins shortly after the singing begins.

The temperature of ebullition, or the *boiling-point*, increases with the pressure,[1] and for a given pressure in any one liquid the boiling-point is always the same. The boiling-point remains unchanged, whether the heat is applied either quickly or slowly or with great intensity. The boiling-point of water under different pressures has been determined by several series of experiments, principally by Regnault, and these are shown in column 2 of the table at the end of Chap. II.

Properties of Vapours.—Atmospheric air particularly and gases generally have the property of absorbing vapours given off from liquids at temperatures below the respective boiling-points of such liquids. The air or gas into which a vapour escapes from the surface of a liquid is not capable of holding up more than a certain quantity, and when it has taken up this quantity the air or gas is said to be saturated with the vapour of the liquid. Thus the atmosphere, when only partially saturated with water vapour, will continue to drink up water vapour at any ordinary temperature either at or below that of the boiling-point. This process is continually going on in nature, and moisture is absorbed from everything which yields it—earth, plants, and animals. Generally, the proportion of vapour required to produce saturation increases as the temperature of the absorbing gas or air increases; and the rate of absorption increases with the temperature at which the vapour is formed. Any decrease in temperature of the absorbing gas or air, when saturated, produces a condensation of vapour which falls in beads of moisture to the earth or other substance beneath.

Evaporation, or formation of vapour, below the boiling-point is very common with alcohol, spirituous oils, and petroleum oils; the particles thus escaping from the surface of the liquid exist in a free state as vapour. Dry air will thus

[1] When the surface of the water is open to the ordinary atmospheric pressure of 14·7 lb. per square inch, fresh water boils at 212° Fahr.; but if the vessel be entirely closed, so that the steam pressure increases, the boiling-point rises, and at about 21 atmospheres' pressure (=300 lb. per square inch), such as now used in Naval boilers, the boiling-point is about 422° Fahr. The atmospheric pressure decreases generally with increased height above sea-level, and water then boils at a lower temperature; thus on top of Mont Blanc (height 15,000 feet) it is at about 183° Fahr. The "boiling-point" also increases with the density of the liquid; thus ordinary sea water, which is of greater density than fresh water, boils at 213·2° Fahr.—not 212° Fahr.—under normal atmospheric pressure.

take up, and retain until saturated, the vapours from petrol motor spirits of about double the quantity at a temperature of 80° Fahr. than at 50° Fahr.; and with certain pressures and proportions of volume, in a closed space the mixtures thus produced become explosive.

Above a certain *critical temperature*, which is peculiar to each vapour and corresponding generally to its chemical composition, the *vapour* becomes a *gas*, and cannot be condensed by *compression alone*. At the critical point the specific volumes of a liquid and of its saturated vapour are equal. This critical temperature for steam and water vapour is above that usually obtained in steam-engine practice, and need not be considered here; but bearing it in mind, it may be stated that in practice any compression of the saturated vapour of water (saturated steam) will thus produce a condensation of some of the particles of which it is composed. Water, or any liquid, contained in a closed vessel and only partially filling it, will give off sufficient vapour to fill the remaining space within the vessel from which the air has been wholly or partially withdrawn.

Vapours are also given off by some solid bodies without apparently passing through the intermediate liquid state. Thus ice gives off vapour, which is absorbed by the atmosphere; also coal, especially when of a gaseous or bituminous nature, gives off hydrocarbon gases which, under certain conditions of temperature and proportionate mixture with the oxygen of the air, become spontaneously ignited and sometimes explosive. (See also "Spontaneous Ignition," Chap. X.)

Transference of Heat.—Heat is transferred from one body to another in three ways—by conduction, by convection, and by radiation; either together, severally, or separately. In any case, heat can only be transferred from a hotter to a colder body.

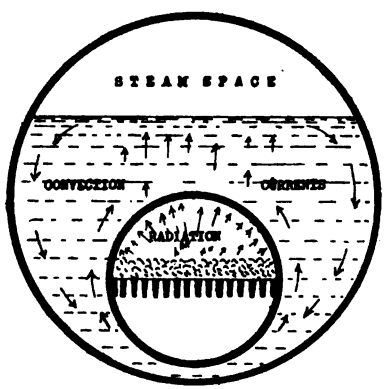


FIG. 13.—Transference of Heat.

In a boiler, for example, as shown in Fig. 13, the fire radiates heat to the furnace and some parts of the combustion chamber; heat then passes through the plating by conduction; and the water is heated and steam formed by convection. The natural draught up the chimney or funnel is caused by convection, and thus draws in the air necessary for combustion from the ashpit.

Conduction occurs when heat passes from one part of a body to another part of the same body, when it may be termed a case of *internal* conduction; or it may occur when heat passes from one body to another in contact with the first, when it may be termed a

case of *external* conduction. The rate of conduction is always greater in internal than in external conduction, principally because it is impossible to obtain perfect contact between two bodies. The nearest approach to perfect contact is probably in alloys of metals, when the rate of conduction is always less than that of the internal rate of the better-conducting metal separately.

Many substances conduct heat very slowly, and are then called *non-conductors* of heat; gases, fluids, earthy substances (such as brick, porcelain, marble, and porous substances generally), are bad conductors, or relatively non-conductors of heat. Steam pipes, cylinders, exteriors of boilers, etc., which should be kept warm, are usually covered with some non-conducting material, commonly called *lagging*, so that as little heat as possible shall escape. For the same reason, bodies which should be kept cool, such as ice, are also covered to prevent heat passing into them from outside influences. In all steam or heat plants the lagging is of great importance, and great attention is paid to its proper provision and repair, as, quite apart from possible inconvenience to persons near, the efficiency of the lagging has an important bearing on the economical results which may be obtained from the engine and plant generally.

Convection.—In fluids and gases heat is transferred internally by convection currents, which practically means a circulation of the heated particles upwards, some heat being transferred to the particles in the first instance by either conduction or radiation. As nearly all liquids and gases expand when heated, the heated particles become lighter than the colder ones, per unit of volume, and their specific gravity becomes less; these lighter particles rise, and their place is taken by heavier and colder ones until the general mass of the fluid becomes heated by the continuous circulation of the convection currents. The application of heat to the upper surface of a fluid merely raises the temperature of the fluid at the surface, and practically no convection takes place; and thus no change of temperature takes place in the lower strata of the fluid. Therefore when a gas or liquid is to be heated, the heat is applied as low down as possible relatively to the fluid, and consequently the furnaces of boilers are generally placed as near the bottom of the boiler as practicable.

For *heating* rooms and buildings, an efficient form of convection heater, shown in Fig. 14, is sometimes used, and consists of a pipe, the body of which is in the chimney space, with its opposite ends opening into the room near the floor at B, and near the ceiling at C. Heat from the ordinary fire in the grate warms the pipe (within the chimney), which heats the air inside the pipe; the hot air rises and

escapes into the upper part of the room, while colder air enters the pipe at the bottom to take its place. When properly arranged on this system, large buildings or rooms can be quickly warmed with a very small fire, or source of heat, without vitiating the air, and with less draughtiness than under the usual conditions.

Ventilation by natural means is always obtained by convection, and to obtain any practical efficiency it is necessary to provide a means of inlet and outlet for the convection currents. The ordinary water-tight or other compartments into which a ship is divided can be most efficiently ventilated by natural means,

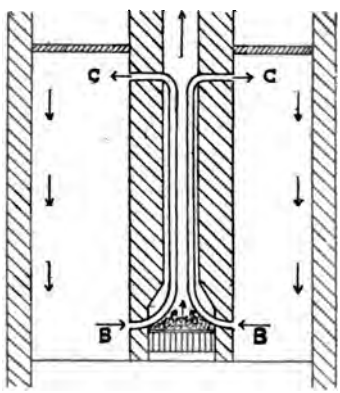


FIG. 14.—Convection Heating (Air).

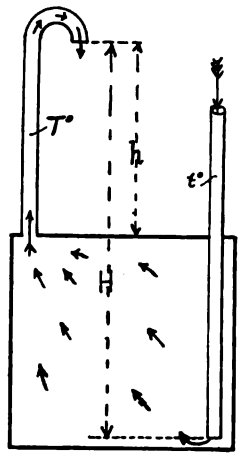


FIG. 15.—Convection (natural) Ventilation.

without the cost of energy or power, by providing an outlet, or uptake, in the roof of the compartment, and an inlet, or downtake, to bring the colder air down to almost the bottom of the compartment, as shown in Fig. 15. If the temperature of the air in the compartment is greater than the external air, the upcast current, set in motion by convection, continuously draws in a cold current through the downtake, and the greater the difference in temperature, the more rapid (increasing with the square root of the difference in temperature) becomes the flow of cold air into the compartment. As the hotter air occupies more space than the colder, the area of the uptake opening should generally be greater than that of the downtake—usually $1\frac{1}{4}$ to $2\frac{1}{2}$ times.

The velocity, or rate of flow, of the air upwards through a

ventilator, or a chimney, is dependent on the head of pressure created by the difference in weight of two columns of ascending and descending air *of the same height as the ascending column.* The only influence which the descending column has on the rate of flow is by virtue of its lower temperature, and to obtain a low temperature it is necessary to tap the supply from a cool place. *Cowls* fitted above supply ventilators may increase the rate of flow by concentrating the air and increasing its density or pressure, thus making it heavier, and this density or pressure is greatest when the temperature is the lowest available. When ventilation is most required—in tropical harbours, for example—cowls are sometimes inoperative, because by introducing the cold air near the top of a compartment, the virtual height of the ascending column is decreased and the air remains stagnant in the lower parts of the compartment.

The velocity (V) in feet per second, due to certain conditions of height (H) of the ascending column, and difference in absolute temperature of the ascending (T_0) and descending column (t_0), is—

$$V = \sqrt{2g \cdot \frac{T_0 - t_0}{t_0} \cdot H},$$

where g = acceleration due to gravity = 32 approximately.

If the absolute temperature of the descending air is about 500° (= 39° Fahr.)—

$$\frac{1}{t_0} = \frac{1}{500} = 0{\cdot}002,$$

which is a value near enough for all ordinary cases of ventilation.

To take an example, if we allow for a rise of temperature of 10° Fahr.—

$$V = \sqrt{64 \times 0{\cdot}002 \times 10 \times H} = 1{\cdot}1\sqrt{H} = \sqrt{H} \text{ nearly}$$

if some allowance is made for frictional losses in the ventilators.

Next, suppose the height of a compartment is 21 feet, $(H - h)$, and the height of the uptake ventilator, h, is 16 feet above this; then the rate of flow is—

= 4 feet per second, when the supply comes in at the top,
or = 6 „ „ „ „ „ „ bottom,

and the rate of flow is considerably increased by any further rise in temperature of the compartment.

A man requires about $\frac{1}{20}$ cubic foot of free air per second, and by a further simple calculation it can be shown that a sufficient supply could be obtained under the above conditions, and neglecting frictional resistances, through a hole only about $\frac{1}{2}$ inch in diameter, the supply coming in near the bottom and the uptake being 16 feet high above the compartment. This hole seems very small, but when a diver's supply pipe is examined it is seen that it is not far from the truth, although the means adopted to ensure the supply differs in method but not in principle; each is caused by a difference in density, or head of pressure — the diver's supply by artificial means, and the compartment by natural means.

The practical difficulties of ventilation generally arise from a misconception of the simple principle involved, and instances are very common where the uptake is practically non-existent (except either in the form of an uncovered hatch, or of a very short bent pipe), and where the supply also enters at the same level; very little natural ventilation can occur under these conditions.

The convection currents which are set up in water when heated are utilised for *heating rooms*, houses, and ships, and in all cases the principle is precisely similar to that in a boiler, although there is necessarily some difference in constructive detail. A simple arrangement is shown in Fig. 16, and is frequently used in the common or garden greenhouse. There is a vertical boiler, B, with an open top or a removable cover, and the fire is lighted either immediately beneath the boiler or in a furnace fitted in its lower part. The convection circulates the water upwards and then away at A, through the pipe C shown, the opposite end of which is brought back to the boiler at the bottom. The hotter water communicates its heat by conduction from the interior to the exterior of the pipe; heat is radiated from the outer surface to the objects in the greenhouse, and the water, thus becoming cooled, returns downwards to the boiler.

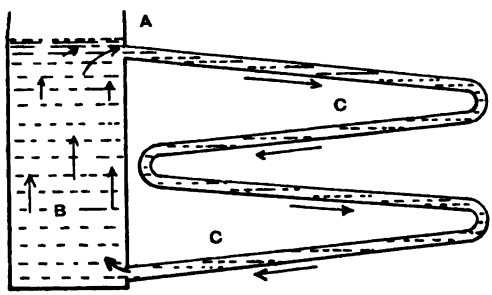


FIG. 16.—Convection Heating (Water).

For heating other places, it is generally necessary to entirely close the boiler so that it is pressure- and water-tight, and to carry the piping vertically upwards to the highest point necessary for the system and place. The boiler and run of pipes must be kept entirely filled with water, and this is done by fitting a cistern or supply tank, with an automatic ball-valve arrangement, above the highest point to which the water can be circulated. An overflow pipe is also necessary,

and is fitted also near the highest point, to prevent overloading or oversupply. A common example of such a system is found in most houses for supplying hot water from a tank fitted in the top of the house, and connected by two pipes, upcast and downcast, with a boiler fitted in rear of the kitchen fire.

In all systems of hot-water heating (as in water-tube boilers also) certain precautions are necessary. The interior of the boiler should be accessible for periodical cleaning; the pipes should incline upwards for the upcast current and downward for the downcast current (vertical, if possible); and near the boiler, where scale is likely to accumulate, the pipes should be vertical and arranged so that the scale, when loosened, should fall into the boiler and not be lodged in the pipes and so choke them. At no part of the pipe system should any part be horizontal, and sufficient inclination should be given so that when the boiler is

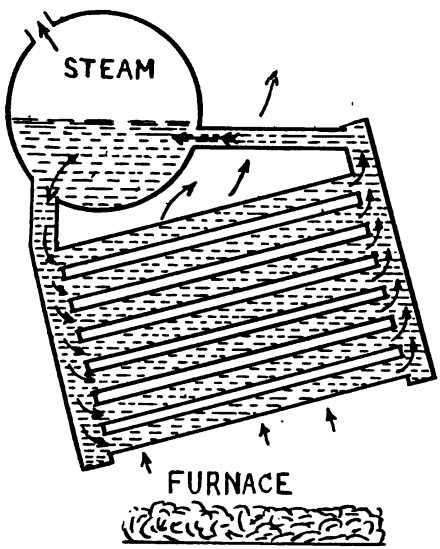


FIG. 17.—Circulation in Boiler.

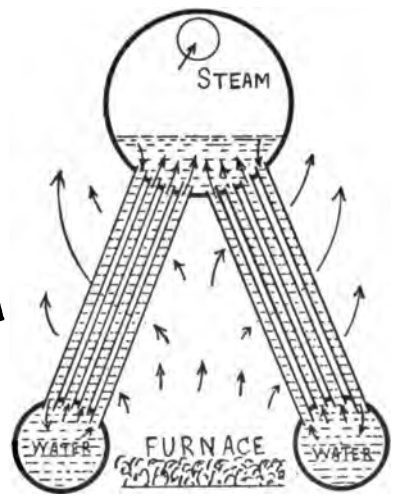


FIG. 18.—Circulation in Boiler.

emptied the whole system is completely drained. If, as on board ship, such complete drainage is difficult (although never impossible), owing to the lack of head room, then drain cocks should be fitted to each pocket so that complete drainage can be obtained in this way. It is evident that circulation by convection in a horizontal pipe cannot occur from this cause alone, and is only obtained from some difference in temperature and level at other parts of the system. If properly designed, even in a ship, self-drainage, without pockets and independent drainage, is quite possible of attainment, and should be the rule and not the exception.

Referring to Fig. 16 it should be noticed that if the fire be applied beneath the tubes, instead of beneath the boiler, the circulation by convection is reversed, and the system becomes virtually a water-tube boiler, in which the original boiler becomes a downtake pipe, as shown in Fig. 17.

A good natural circulation, which can only be obtained by convection, is essential to the safe working of all boilers. In recent years, water-tube boilers

have been gradually developed into two distinct classes: (1) those in which the tubes are nearly horizontally arranged, as shown in Fig. 17; and (2) those in which the tubes are nearly upright as shown in Fig. 18, and in which the circulation is naturally rapid. With a slow circulation, the heat conveyed to the tubes is taken away slowly by the sluggish movement of the water; bubbles of steam, which is practically a non-conductor of heat, are formed, and, not escaping quickly enough, overheating of the material of the tubes is therefore probable when more than a moderate quantity of heat is given out by the fuel from the furnace. A rapid circulation of water, as in a boiler with nearly upright tubes, allows a maximum quantity of fuel to be burnt, and a maximum quantity of steam to be formed, without any tendency to overheat the material; but, as in all boilers, only for so long as the water side of the heating surface remains clean and free from grease and other deposits of scale.

The figures are typical of the direction of circulation by convection, and do not exactly represent any particular boilers. Every boiler is a study in itself, and a conclusion should not be drawn either that the boiler shown in Fig. 17, for example, is necessarily unsafe, or that that shown in Fig. 18 is comparatively a great improvement, without a full consideration of other merits or disadvantages, in conjunction with the proper allowance for unequal expansion of the material, the weight, the space, and the work which the boiler is expected to do.

The temperate climate of the British Isles, and also that of Japan in a lesser degree, is due to convection in conjunction with certain *ocean currents*, the upper surface of which, being always warm, maintains an equable temperature in the shallower water which surrounds the coasts. These ocean currents are caused, in the first instance, by the rotation of the earth as a globe about its axis, pitched nearly at the poles, north and south. The solid part of the world, the land, rotates with uniform angular velocity about this axis, from west to east; but the ocean, being fluid and not solid, does not keep pace exactly with it. Near the equator, where the earth has its greatest velocity (not angular velocity, which is equal from pole to pole), the water fails to keep pace in the greatest degree, and causes a steady and apparent current or flow from east to west, generally called the equatorial current. Near the poles, where the earth has little or no velocity, there is practically no current caused in this way. The direction which the equatorial current takes is generally determined by the contour of the coast-lines, but is also assisted by the varying depth and formation of the bed of the ocean. The surface water of the ocean near the equator is always warmed by the radiant heat of the sun, and this warmer water, tending to always remain near the surface, accounts for these currents continuing to be warm long after they have left the equator and have almost reached the poles.

The Gulf Stream is a notable instance of an ocean current, warm water being carried from the Gulf of Mexico, past the Florida peninsula, northward and eastward towards Iceland and the North Pole, then southward round the British Isles, and is finally lost in the equatorial current. The colder water flows in varying direction beneath the warmer current, and thus tends to maintain a constant level. The colder current sometimes reaches the surface, a particular case being evident in a southward current moving along a narrow belt off the east coast of North America until it apparently plunges below the warmer Gulf Stream.

Heat is imparted from one body to another and colder one *through space* by *radiation*, in exactly the same way that light is radiated from a luminous body. The rays of heat are sent out in every direction in straight lines, invisible to the eye, and the intensity varies very nearly *inversely* as the square of the distance between the two bodies, one giving out and the other receiving heat. It is in this way that a common fire imparts its heat to the walls and other objects in a room, and also in the same way that a boiler fire communicates some of its heat to the furnace plating and sides. A rough body radiates more heat, or receives more radiant heat, than a smooth or polished body (principally because the surface area is greater in rough than in smooth bodies), and therefore a polished kettle or a steam pipe will radiate less heat than one which is rough and unpolished; but the oiling of a smooth surface increases its power of radiation.

The rays of heat received by the earth from the sun are absorbed by the intervening atmosphere to some slight extent, and by moisture in the atmosphere to a greater extent. The intensity of the heat radiated to the earth is diminished by clouds of vapour in the atmosphere and by the relative density of the atmosphere. Vapour rising from the earth is gradually reduced in temperature with the reduced pressure of the atmosphere at greater height (its boiling-point is lowered, but its capacity to hold vapour is also decreased), and, condensation eventually ensuing, beads of moisture are formed. This form of moisture, being of greater weight or density than the mixture of air and vapour, is either re-evaporated at a less height, or deposited as dew, rain, hail, or snow on the earth. A continual cooling of the upper strata of the atmosphere is therefore always taking place, and at great heights, such as the tops of mountains, moisture is deposited at a low temperature, producing, in well-defined instances, perpetual snow.

[TABLE

COMPARATIVE HEAT PROPERTIES OF SOME SUBSTANCES

	Specific Heat.	Radiating or absorbing.	Reflecting.	Conductivity. (k) per cent.		Linear Expansion. Coefficient for 1.
				At 32° F.	At 212°.	
Aluminium. .	0·202	...	...	34·3	36·2	...
Antimony . .	0·051	...	...	44·2	39·6	0·00000617
Brass . . .	...	11 (dead polish)	89	20·4	25·4	0·0000106
Brass . . .	...	7 (bright)	93	...	...	0·0000104
Carbon . .	0·238 at 60° F.	...	...	...	...	...
Carbon . .	0·456 at 1750° F.	...	...	...	...	...
Copper . .	0·092	7	93	71·9	72·2	0·0000101
Iron, cast . .	0·13	25	75	16·6	16·3	0·0000065
Iron, wrought .	0·11 to 0·13	23	77	...	...	0·00000681
Steel . . .	0·116	...	...	...	...	0·00000668
Lead . . .	0·032	...	...	...	...	...
Lampblack (soot)	...	100	...	8·36	7·64	0·0000159
Mercury . .	0·032	...	...	1·48	1·89 at 110° F.	...
Nickel . .	0·108	...	...	...	...	...
Platinum . .	0·32	...	...	...	...	0·00000484
Silver. . .	0·56	3	97	96	...	0·0000112
Sulphur . .	0·184	...	...	...	...	...
Tin . . .	0·56	...	...	15·28	14·23	0·0000132
Zinc . . .	0·094	...	...	30·3	...	0·0000173
Ice at 32° F. .	0·504	...	...	...	...	0·0000286 from −17° to 30° F.
Water at 39·1° F.	1·00000	100	...	0·136%	...	...
Water at 212° F.	1·013	...	...	...	...	...

Mechanical Equivalent of Heat.—The following laws (of thermodynamics) are sufficiently true for all practical purposes, and for very close investigation, and are the results of many careful and exhaustive experiments by Rumford, Joule, and others:—

First Law.—When mechanical energy is produced from heat, a definite quantity of heat goes out of existence for every unit of work done; and conversely, when heat is produced by the expenditure of mechanical energy, the same definite quantity of heat comes into existence for every unit of work spent. This may be written—

Heat supplied = gain in internal energy + work done.

In other words, the energy expended in raising the temperature of 1 lb. by weight of water 1° (Fahr.) is capable of doing 780 foot-lb. of mechanical work; and conversely, by expending 780 foot-lb. of work, the temperature of 1 lb. by weight of water can be raised 1°: one British Thermal Unit being then produced. We then say that the "mechanical equivalent of heat" is 780 foot-lb., or—

1 British Thermal Unit = 780 foot-lb.[1];

and is commonly known as Joule's Equivalent = J = 780.

Second Law.—It is impossible for a self-acting machine, unaided by any external agency, to convey heat from one body to another at a higher temperature.

In accordance with the first law, it would be possible for the whole of the heat energy in a substance to be converted into mechanical energy, but the second law sets a limit on the quantity which can be so converted, and a certain quantity is left unconverted and is necessarily "rejected" as heat. Consequently, the efficiency of the conversion, or of the machine effecting the conversion, must be less than unity, and $= \frac{\text{heat converted into work}}{\text{heat taken in by the engine}}$.

In Chap. II. *work* and *energy* were defined, and the unit of measurement was stated as 1 foot-lb. If expressed in heat units, this unit of 1 foot-lb. becomes $\frac{1}{780}$ of 1 British Thermal Unit. It should be clearly understood that no work can be done by an engine without a corresponding fall in pressure of the air, gas, or liquid used to produce motion. This fall in pressure must always be accompanied by a corresponding fall in *temperature.* In all cases the internal energy, or stock of energy, contained in the substance is decreased by exactly an equal definite quantity as the work done.

Energy cannot be dissipated or lost. If all the available energy contained in a substance apparently disappears, and only some part of it is converted into work, then the remainder has been transformed into some other form of energy, and still exists in this other form. Commonly, the energy thus apparently wasted disappears as heat, which is usually produced by friction and diffused into the atmosphere or other surroundings.

The law of the **Conservation of Energy** states: "That the total energy of any one form to which the energy of a given system

[1] Various experiments by Joule, 1849-78, made this value 772-774
,, ,, Rowland, 1879, ,, 778
,, ,, Schuster, 1894, ,, 775
,, ,, Rowland, 1895, ,, 778·3
,, ,, Osborne-Reynolds, 1897, ,, 777

The British Association recommended, in 1896, the adoption of 780.

The number 780 is a very convenient one, being divisible by 2, 3, 4, 5, 6, and 13, and by various multiples of these numbers.

of bodies is reducible, is unalterable so long as the system is not acted on from without." This law is not only true of mechanical energy, but of all kinds of energy, such as those of heat and electricity. In all that follows relating to the conversion or transference of energy from one form to another form, this law is the fundamental basis, and should be thoroughly understood.

For example:—Potential energy is supplied to a ship in the form of coal or other fuel. Heat is generated by combustion of the fuel, and some of it is converted by the engine into mechanical energy which is used in rotating the propeller. There are various losses (briefly summarised under the heading of *Efficiency*) during the various processes of conversion, and, finally, the whole of the potential energy originally contained in the fuel is returned to the earth as heat energy.

The **Natural Sources of Energy** are many, and in nearly all cases the sun is the original source, energy being derived by its action on various natural substances. Thus the heat from the sun produces—

1. *Wind energy*, derived from the convection currents in the atmosphere, producing varying densities, with difference of pressure or force per unit of area. The colder air moves along near the earth's surface, to take the place of the hotter and ascending air, and produces wind.

2. *Water energy*, derived from evaporation of water, the vapour of which, mingling with the atmosphere, forms clouds, from which rain, etc., is deposited on the earth. Water deposited on heights above the ordinary sea-level possesses a certain potential energy, and when flowing to the lower level this energy can be utilised in driving water turbines or other water-driven motors.

3. *Heat energy*, derived from certain decompositions of matter, resulting in the formation of coal, oil, and other fuels, which, when burnt, develop energy by combustion in the form of heat.

4. *Food energy* (actually a form of heat energy), from the growth of plants and animals.

Other sources of energy are—

5. *Tidal energy*, considered to be principally derived from the moon's attraction; and

6. *Electrical energy*, which exists in nature, but has very little value at present for practical purposes, and which is probably some variety of heat energy.

Electrical energy in common use is *not* derived from a natural source, but is obtained by utilising other forms of energy to produce it; it is therefore a secondary motive power. Similarly, hydraulic engines when worked by pressure produced, not from a natural source, but by utilising some other form of energy, are secondary motors. Electricity and hydraulic pressure are generally used as mediums for transmitting energy from one place to another, and should therefore usually be considered, not as motive powers, but as a means of transmission of energy; thus taking the place of some purely mechanical transmission.

Combustion is the chemical union of a combustible or fuel, such as coal, coke, petroleum, naphtha, kerosene, benzine, gasolene, alcohol, and petrol or other spirit, with oxygen; combustion takes place rapidly only at a high temperature. When such fuel is burnt, large volumes of gas are given off, which, if confined to a small space, allow great pressures to also be produced. From combustion the heat energy utilised in all primary heat engines (steam engines, guns, oil and gas engines particularly) is derived.

For the ordinary steam engine, heat is supplied by the combustion of coal, oil, or other fuel in the furnace, and steam is generated in the boiler. The heat energy contained in the steam thus produced is utilised, as far as possible, in doing work through the medium of the engine; but some part of the energy is not utilised, and is then termed "heat rejected."

For oil and gas engines there is no boiler, and the combustible, oil, spirit, or gas, is injected into the cylinder itself, with a certain proportionate mixture of air, which contains the oxygen necessary for combustion. The mixture is ignited and burnt into gases; a high temperature is produced, and a correspondingly high pressure, which, acting on the piston, produces motion and does work.

For guns, certain chemical substances, containing sufficient oxygen for more or less complete combustion, are ignited by detonation, and combustion follows with a rapid formation of gases at a high temperature and a correspondingly very high pressure. This pressure sets the projectile in motion and does work in exact proportion to the heat energy expended within the bore of the gun.

Internal ballistics, which is the name generally given to the science of heat applied to guns, is neither more nor less than the science of heat applied to the steam or other heat engine.

Heat engines—in which the combustion takes place within the cylinder—are known as *Internal Combustion Engines.*

In a compressed air engine the energy used in the original compression of the air is transferred as heat to the air so compressed. This heat is expended by the compressed air when it is used to do work on some external body, and in doing so the air loses heat energy in exact proportion to the work done or resistance overcome in doing the work.

CHAPTER IV

FORMATION AND EXPANSION OF STEAM

Formation of Steam.—In a closed vessel, such as a boiler, containing partly air or steam and partly water, steam is produced by applying heat to the water. As the temperature rises, more and more vapour is given out from the water until at a certain temperature, called the boiling-point, ebullition begins. The steam as it is formed has only about the same volumetric space to fill, and gradually becomes compressed within the steam space above the water level; the pressure of the steam thus rises as more and more steam is generated. As the pressure rises the boiling-point also rises in conjunction with it, but at a certain pressure known as the working pressure, no further rise takes place because some of the steam is either conducted away through a pipe to the engine, or escapes through the safety valves which do not allow any further accumulation of pressure and limit it to a safe working load.

The steam escapes regularly or irregularly from the water surface according to the purity or impurity of the water. Only under very peculiar circumstances does each unit of surface yield an equal volume of steam. In an ordinary marine boiler steam is generally given off at some prominent point, perhaps a stay or rivet head, but the total area of the water surface plays its part in steam formation by supplying the heat necessary for evaporation and for breaking the cohesion of the molecules of water and converting them into steam. The evolution of the steam supplies the necessary circulation and movement of the water towards the points at which steam is given off.

In the process of raising steam and of afterwards maintaining a constant working pressure in a boiler, two changes must be considered—viz. the change of temperature necessary to bring the

water up to the boiling-point, and the change of state necessary to convert the water into steam at the boiling-point.

Quality of Steam.—The steam formed in a boiler may be of two qualities, either *saturated* or *humid* (or *wet*), and wet steam contains various proportions of moisture or particles of water in suspension.

When steam contains no moisture, and at the same time the temperature is not raised above that of the boiling-point due to its pressure, it is said to be saturated, or dry *saturated steam.* The term *saturated* means that the steam is saturated with heat because evaporation is complete.

In general practice, although both the temperature of the boiling-point and the pressure correspond to those of saturated steam (as shown on p. 18), there is some moisture in suspension, and the steam is humid or wet; it is then called either *common steam,* if only a small proportion of moisture (3 to 5 per cent by weight) is present in it on leaving the boiler, or *surcharged* steam, if the proportion of moisture is large. *Priming* is said to take place when an undue proportion of moisture or water in suspension is carried off by the steam in its passage from the boiler.

When heat is added to saturated steam so that, all the moisture being then evaporated, the temperature of the steam is raised above that of the boiling-point due to its pressure, it is termed *superheated steam.* Steam cannot be superheated in the boiler in which it is generated under ordinary conditions of working, but if the water-level is allowed to fall below that of the heating surface, it is possible to superheat the steam; but this is not a probable result or occurrence. When superheated steam is required the extra heat is added in a separate vessel to that in which the steam is formed, so that it shall not be in contact with the water from which it is generated.

In the ordinary process of producing superheated from common steam, the moisture is considered to be first converted into saturated steam (with no rise of temperature until the whole mass is saturated). As superheated steam, the temperature rises in a manner almost identical with a gas similarly heated. The specific heat of superheated steam near the boiling-point is about 0·48, and therefore for each b.t. unit of heat added to each 1 lb. by weight of dry steam a rise in temperature of about 2° Fahr. takes place in the superheated steam. Largely varying values have been found by experiment for the specific heat of steam at various pressures and degrees of superheat. It is highly probable that some moisture is always present, even when the steam is considerably super-

heated; although this probability should not be overlooked, the generally accepted view, "that no moisture is present as soon as superheat begins to make its presence distinctly known" (by showing a temperature above that of the boiling-point due to its pressure), is sufficiently accurate for ordinary purposes.

The relative values of *pressure* and *temperature* of saturated steam are given in the Table of Properties of Saturated Steam at the end of Chap. II., which is based on experiment. The temperature rises with the pressure, but not in direct proportion as does a gas like air, and the tabulated results are nearly always used by engineers. Some empirical formulæ have, however, been derived from the table and used for instructional purposes, but they are of little practical use.

Total Heat of Formation of Steam.—From Regnault's and other experiments it is found that the latent heat of evaporation (see column 7; p. 18) decreases with an increase of pressure and temperature, according to the approximate formula—

$$\text{Latent heat, } L = 966 - 0{\cdot}7(T - 212),$$

where 966 = latent heat of evaporation of 1 lb. of water under atmospheric pressure; and T = the temperature of evaporation, or boiling-point.

If t = temperature of the water before heat is applied, then sensible heat, $S = T - t$ (see column 6, p. 18).

From the above, the *total heat of evaporation* of 1 lb. of water into saturated steam—

$$\begin{aligned} H &= L + S \\ &= 966 - 0{\cdot}7(T - 212) + (T - t). \end{aligned}$$

This formula shows, as well as the table, that the total heat of evaporation of saturated steam gradually increases, although the latent heat decreases, with the pressure and temperature.

Evaporation is generally considered to be *complete* evaporation of the liquid into gas. In the *formation* of steam some moisture is generally present in the steam, and evaporation is thus *incomplete*; and we thus speak of the "total heat of formation" of such steam, not of the "total heat of evaporation."

The proportion of dry steam, by weight, contained in each 1 lb., by weight, of common steam is called the *dryness fraction* of that steam. As stated above, there is always some moisture in common steam produced in a boiler; the whole of each 1 lb. by weight of

water is, however, raised to the boiling-point, and the sensible heat $(T-t)$ remains unchanged; yet only a fraction or percentage of this 1 lb. may be converted into saturated steam. Let this fraction $= x =$ dryness fraction, and then $(1-x)$ is the fraction by weight of moisture present in the steam; then the latent heat expended in producing each 1 lb. of common steam is

$$= x \,.\, L$$
$$= (x) \,.\, [966 - 0{\cdot}7(T - 212)],$$

and the total heat of formation $= x \,.\, L + S$

$$= (x) \,.\, [966 - 0{\cdot}7(T - 212)] + (T - t).$$

It should always be remembered that when heat is taken away from the steam and it is reconverted into water, that exactly the same quantity of heat is given up during the reconversion as was required originally to convert it into steam. Thus when steam is condensed to water (in the ordinary process of condensation, by adding water to it or by spraying the steam on to some cold surface), the steam gives up heat in exactly the same quantity as would be required to raise it from its state as water (to which it is reduced, and at the same temperature) to the temperature of the steam from which it is reduced, and to the same state of saturation, or condition of common steam.

EXAMPLE.—Find the quantity of heat given up by 1 lb. weight of steam at 295 lb. gauge pressure, and superheated to 620° Fahr., when reduced to ice at 0° Fahr. and normal atmospheric pressure.

		B.t. units.
Change of temperature,	sensible heat, from 620° to 420° = 200°, at 0·48 b.t. units per 1° F.	= 96
Change of state,	latent heat of evaporation, at 420° F.	= 815
Change of temperature,	sensible heat, from 420° to 32° F.	= 388
Change of state,	latent heat of fusion, at 32° F.	= 144
Change of temperature,	sensible heat, from 32° to 0° = 32°, at 0·5 b.t. units per 1° F.	= 16
	Total	= 1459

Some difference must exist between the pressure of the steam as it leaves the boiler and its pressure on arriving at the engine, or no motion or movement of the steam inside the pipe could take place. A fall in pressure must be accompanied by a fall in temperature. Again, there is another difference of pressure between that in the pipe near the engine, and that on the engine side of the valve which is used to regulate the supply of steam to the engine; at low powers this latter difference is very great, while the former is decreased. Whenever motion is produced, work is done and energy expended; consequently, some loss must take place in the internal heat energy of the steam in its passage through the steam pipe and through the regulating valve, producing a tendency to condense some of the steam to water, and therefore to produce a greater percentage of moisture in the steam on its arrival at the engine. This tendency to condensation is counteracted to some extent by the fall in pressure and temperature, which tends to release some of the heat necessary to keep the steam in a state of saturation or other original condition, because steam of lower

pressure requires less total heat per 1 lb. than steam of the initial or higher pressure. If the steam leaves the boiler as saturated steam, seldom attained in practice, there is a slight tendency to produce a condition of superheating, because the steam may eventually possess a greater temperature than that naturally due to its pressure. The same remark applies to steam with little moisture ; but in all cases, in addition to the losses of heat in doing work (in producing motion of the piston and of the steam itself), there is a comparatively large loss by radiation of heat from the exteriors of pipes and valves, which prevents, or tends to prevent, any superheating of steam by a fall of pressure alone, in practice.

Expansion Curves.—Steam is not a perfect gas, and does not expand exactly in accordance with Boyle's law—P . V = constant—but somewhat differently, according to the conditions under which the engine is working. In Fig. 19 three curves are shown—

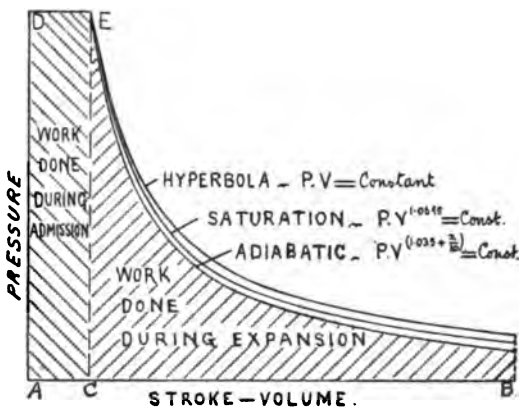


FIG. 19.—Expansion Curves.

1. The *hyperbola*, which corresponds with Boyle's law.

2. The *saturation curve*, which is obtained by plotting a curve of pressures and corresponding volumes as shown in the Table for Saturated Steam. In this case, at any instant during the expansion, any increase of pressure will produce condensation of some of the steam, while any decrease of pressure tends to produce superheating :

$$P \cdot V^{1 \cdot 0645} = \text{const.}$$

3. An *adiabatic curve*, when steam expands in a cylinder without either taking in heat from, or giving out heat to, external bodies. Work is done by the steam in this case either by pushing a piston before it or by producing motion in itself, and thus creating kinetic energy :

$$P \cdot V^{1 \cdot 035 + \frac{x}{10}} = \text{const.}$$

The above formula for adiabatic expansion of steam is attributable to Zeuner, and usually covers the range of moisture found in

marine steam engine practice; its limit of usefulness is reached when the moisture exceeds 30 per cent, *i.e.* when $x = 0{\cdot}7$.

If the steam be initially dry saturated, then $x = 1$,

and $$P \cdot V^{1{\cdot}135} = \text{const.}$$

If the steam initially contains 20 per cent of moisture, then $x = 0{\cdot}8$,

and $$P \cdot V^{1{\cdot}115} = \text{const.}$$

which may be written (approximately)

$$P \cdot V^{1\frac{1}{9}} = \text{const.}$$

Hyperbolic Expansion of Steam.—For rough practical purposes the hyperbolic law of expansion holds good for steam as the working substance in the ordinary reciprocating (piston) engine; but, unlike the law as applied to gases, the expansion of steam is in all cases accompanied by a corresponding fall in temperature, and therefore the temperature is not constant.

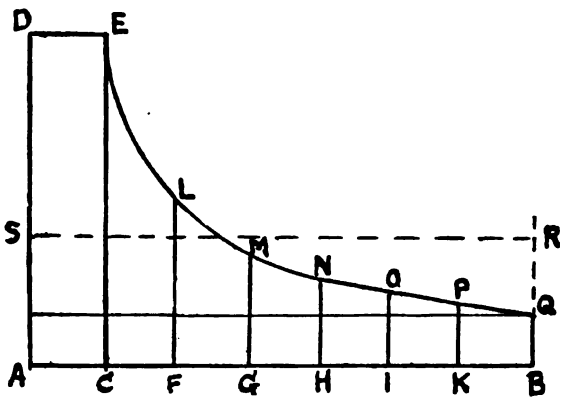


FIG. 20.—Hyperbolic Expansion.

For rough practical purposes, etc., steam, like all vapours, tends to expand infinitely, and on this property the economical working of the steam engine depends.

In Fig. 20, let AB represent the stroke of the piston to any scale, and let AD represent the pressure in the cylinder at the commencement of the stroke. AD is at right angles to AB.

Suppose the steam to act with uniform pressure throughout AC, a fractional part of AB, and that at a point, E, steam is cut off, $CE = AD$, $DE = AC$; then at a point F, $CF = AC$, the pressure will fall to half that at E; represent this by $FL = \frac{1}{2}CE$ or $\frac{1}{2}AD$, and so on.

		lb.			lb.
Pressure at	C is CE = say,	210	Mean, from A to	C =	210·0
„	F is $FL = \frac{1}{2}CE$ =	105	„ „	C to F =	157·5
„	G is $GM = \frac{1}{3}CE$ =	70	„ „	F to G =	87·5
„	H is $HN = \frac{1}{4}CE$ =	52·5	„ „	G to H =	61·25
„	I is $IO = \frac{1}{5}E$ =	42	„ „	H to I =	47·25
„	K is $KP = \frac{1}{6}CE$ =	35	„ „	I to K =	38·5
„	B is $BQ = \frac{1}{7}CE$ =	30	„ „	K to B =	32·5
				Total =	634·50
			Mean of means, $\frac{1}{7}$ =		90·64

If a free curve be drawn through the points E, L, M, N, O, P, Q, this curve is a hyperbola, and it then represents the pressure at any point of the stroke graphically to scale.

The *work done* in the cylinder is represented to scale by the area *DENQBA* of this figure, for it represents both the pressure on the piston and the length of the stroke. (The line AB is taken to represent the back or exhaust pressure.)

The work done on the piston may be divided into two parts—

1. The work done *during admission*, in which the steam pressure is retained at approximately its full or initial pressure, as represented by the area *ADEC*.

2. The work done *during expansion*, in which, after the supply is cut off and no more steam is admitted, work continues to be done, as represented by the area *CENQB*.

Figs. 11 and 19 show how *increased work* and economy of steam can be obtained from the same weight of steam by its *increased expansion*. By increasing the length of the cylinder and stroke of the piston the steam would continue to do work during its further expansion, although the amount admitted would remain the same. The reduction in pressure, obtained by increased expansion, is practically limited in piston engines to the pressure required to overcome the *back* pressure on the piston and the external resistance. There are also other practical limits exercised by considerations of weight and space.

When further admission of steam is cut off, say, at C; the ratio which AB, the full stroke, bears to AC, that part of the stroke before cut-off, is termed the *ratio of expansion* of the steam: it is generally denoted by r, and the cut-off then takes place at $\frac{1}{r}$ of the

stroke. It is evident that the greater r is, the greater becomes the quantity of work which can be obtained from each 1 lb., by weight, of steam used in the cylinder; and thus steam is economised and greater efficiency obtained by increased expansion.

Economy by Increased Pressure is obtained by a possibility of increased expansion. Thus exhausting into the condenser, at 10 lb. absolute, the ratio of expansion with 150 lb. pressure is limited to $= \frac{150}{10} = 15$, and the ratio of expansion with 250 lb. pressure is limited to $= \frac{250}{10} = 25$. The generation of steam of 250 lb. pressure costs very slightly more heat than the generation of steam at 150 lb. pressure, as shown by the table at the end of Chap. II., viz. as 1203·7 is to 1190·7 b.t. units.

Useful Work done by 1 lb. (weight) of Steam.—By constructing a figure, as shown in Fig. 19, to scale to represent graphically the absolute pressure P and the volume V of 1 lb. weight of steam, the quantity of work which it can do may be measured from the area of the figure. For purposes of comparison it is usual to consider the cylinder, or piston, to be 1 square foot in area, so that its length up to the point of cut-off, or the stroke volume up to that point, is equal to the specific volume V of the steam at the absolute pressure P.

The work done during admission = P.V, and if the hyperbolic law holds good for steam, this is a *constant* quantity, and the work done by each 1 lb. weight of steam during admission is always the same whatever the pressure may be.

Similarly, the work done during expansion is also constant, if the same ratio of expansion be adopted, whatever the original pressure may be. But the drawback in practice is, that the lower pressure requires a larger cylinder in which to perform the work, varying directly with the absolute pressure.

For example, 1 lb. weight of steam at 60 lb. absolute pressure fills *twice* the volume of that at 120 lb. pressure; and if the ratio of expansion be 4 in each case, the mean pressure during both admission and expansion respectively—

= 38 and 76 nearly. (Using the graphical means of calculation given above.)

Total work done by 1 lb. weight of steam
- = 38 × 2.V; with 60 lb. admission pressure (absolute)
- = 76.V; with 120 lb. admission pressure (absolute)
- = constant: for same ratio of expansion, and assuming hyperbolic expansion.

Equivalent Evaporation from and at 212° F.—The actual weight of water converted into steam by 1 lb. of fuel varies with the pressure and temperature at which it is formed, and with the temperature at which the feed water enters the boiler. Therefore, although the same quantity of heat may be usefully expended in making steam, the actual weight of steam produced would not be a fair measure for the comparison of evaporation per 1 lb. of fuel. A standard quantity of heat, 966 b.t. units, the quantity necessary to convert 1 lb. of water into steam at 212° F. under normal atmospheric pressure, and when the feed water is at the same temperature, 212° F., is used by engineers for purposes of comparison. If the actual weight of steam produced per 1 lb. of fuel be n . lb., and the heat required per 1 lb. weight of steam $= x$. L + S, then the equivalent evaporation from and at 212° F.—

$$= \frac{n \,.\, (x \,.\, \mathrm{L} + \mathrm{S})}{966}.$$

The expression, $\frac{x \,.\, \mathrm{L} + \mathrm{S}}{966}$, is generally called the "factor of evaporation." When the steam is dry saturated (*i.e.* when $x =$ dryness fraction $= 1$)—

$$\text{equivalent evaporation from and at } 212° \text{ F.} = \frac{n \,.\, (\mathrm{L} + \mathrm{S})}{966}.$$

For *superheated steam*, taking the average value of the specific heat at 0·48 b.t. units, and the dryness fraction = 1, the

$$\text{equivalent evaporation} = \frac{n \,.\, (\mathrm{L} + \mathrm{S}) + (\mathrm{T}_s - \mathrm{T}) \times 0{\cdot}48}{966}$$

where T_s is the higher temperature of the superheated steam.

CHAPTER V

MEASUREMENT OF POWER

Measurement of Pressure.—Work can be represented by an area, and this is also true of energy. Both work and energy are the products of two factors—*Force* in lb., and *Distance* in feet—which can be represented by the area of a plane figure, which is

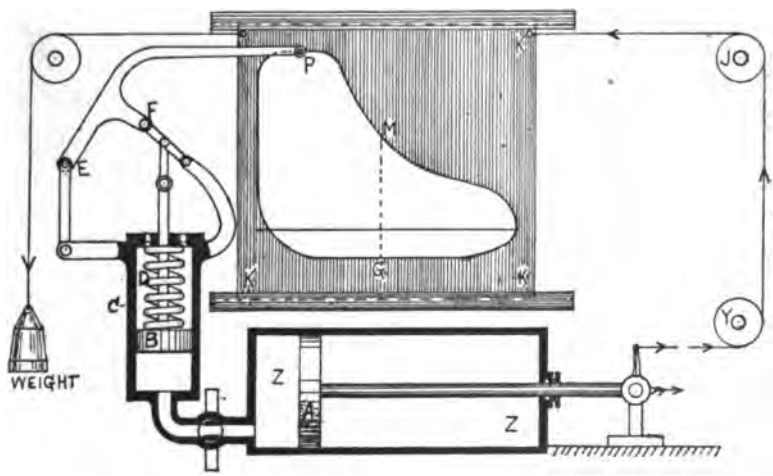


FIG. 21.—Principle of the Indicator.

also the product of two factors. This is practically used for measuring the work done in engines by graphical means with the assistance of the *indicator*.

A figure, such as is marked in Fig. 21, is obtained in practice by the use of the **Indicator** (more fully explained in a later chapter), whose general principle will now be examined and described.

In Fig. 21 there is a cylinder, *ZZ*, in which a piston, *A*, works.

At one end of the cylinder there is a short pipe connecting it with the space below a small piston, *B*, which works vertically in a cylinder, *C*. Above *B* there is a spring, *D*, which presses the piston downwards while the pressure below the piston pushes it upwards. The result of these two actions on the piston is that every inch of movement given to it, by change in pressure, represents a definite difference in the rise or fall of the pressure itself. To this piston, *B*, a rod and a connecting rod are attached, so that the movement of the piston also moves a lever, *EFP*, about its fulcrum *E*. A pencil, *P*, is attached to the lever at *P*, which can be pressed lightly on a piece of paper stretched on a board, *KKK*; thus as the pressure in the cylinder *ZZ* rises or falls, the pencil *P* also moves up or down, and makes a mark on the paper corresponding with the pressure to scale according to the proportions of the lever *EFP* and the strength of the indicator spring *D* which is in use.

The steam piston-rod has attached to it, in some simple way, a string, running over loose pulleys *Y* and *J*, which is attached to a board or flat piece of metal, *KK*, which can move easily to and fro in a slide. At the opposite end of *K* there is another loose pulley and string, and a weight which always keeps the string tight, so that when the piston *A* moves to the right then *K* moves to the left, and *vice versa*. To *KKK* the paper or indicator card is attached, and if the pencil is pressed lightly on the card while the piston makes two complete working strokes, one forward and one return stroke, a figure (shown) is marked by the pencil. This figure represents the pressure on the piston in the end of the cylinder connected with the indicator for the two complete strokes. On the forward stroke the pressure is represented by the upper bounding line of the figure, and on the return stroke by the lower bounding line, which is nearly horizontal. During the return stroke the back of the piston is only subject to exhaust or back pressure.

To obtain the **mean forward pressure,** a line must be drawn horizontally on the card, which must be to the proper scale to represent the atmospheric pressure. The top end of the indicator cylinder is always open to the atmosphere, and by connecting the bottom of the cylinder below the indicator piston with the atmospheric pressure (a small cock is fitted in the short pipe between *C* and *Z* for this purpose), the spring attains its natural

length and is inoperative. Then by moving the card by hand or otherwise for the length of the stroke, a horizontal line representing the atmospheric pressure is marked by the pencil. By finding the mean height of the upper line of the figure above this atmospheric line we obtain the mean forward pressure. Also by finding the mean height of the lower line above, or below, the atmospheric line we obtain the **mean back pressure** on the piston. From what has been stated above it will be seen that the—

$$\text{Mean } \textit{effective} \text{ pressure} = \begin{Bmatrix} \text{mean } \textit{forward} \\ \text{absolute pressure} \end{Bmatrix} - \begin{Bmatrix} \text{mean } \textit{back} \\ \text{absolute pressure.} \end{Bmatrix}$$

A similar diagram is obtained for the other end of the steam cylinder, and the mean pressure on the front of the piston can be obtained for the return stroke in a similar way.

Calculation of Mean Pressure.—The mean pressure is obtained from the indicator diagrams taken at opposite ends of the cylinder. There is a supposition that the figures obtained from the opposite ends are exactly similar, and the forward pressure at one end is usually taken in conjunction with the back pressure on the return stroke at the same end. The actual effective pressure on the piston at any point is, however, equal to the forward pressure *less* the back pressure on the opposite side of the piston. At a certain point, not far from the end of the stroke, the back pressure is greater than the forward pressure, and the inertia of the engine receives no effective assistance, but has to overcome a *negative* effective pressure.

Consider the pair of diagrams shown in Fig. 22—

The work done from left to right = area $ABKC$ − area MCE,
and from right to left = area $DEKF$ − area NFB.

Add together: then total work done per revolution, two strokes—

= areas $(ANFL + FKCL + BFK + NFB - MCE) + (DMCL + FKCL + CKE - NFB + MCE)$
= area $(ANFL + FKCL + CKE) + (DMCL + FKCL + BFK)$
= area $ANFKECL$ + area $DMCKBFL$
= areas contained within the diagrams measured separately.

The *work done*, of which the indicator diagram gives an exact measure to scale, is represented by the area of the two figures contained within the lines drawn for each figure separately. Any

variation of the exact similarity of the two figures produces a difference in the average turning-moment during the forward and return strokes. The examination of Fig. 259, an extreme case, shows such variation. Although the sum of the areas of the two figures represents accurately the work done in one revolution, the variation of turning-moment on the forward and return strokes is very great. In considering the design of an engine, the calculation of turning-moments must be based on the actual difference of pressure on the two sides of the piston, not simply on the mean pressure obtainable from the diagram.

The diagrams are divided horizontally into ten equal parts by

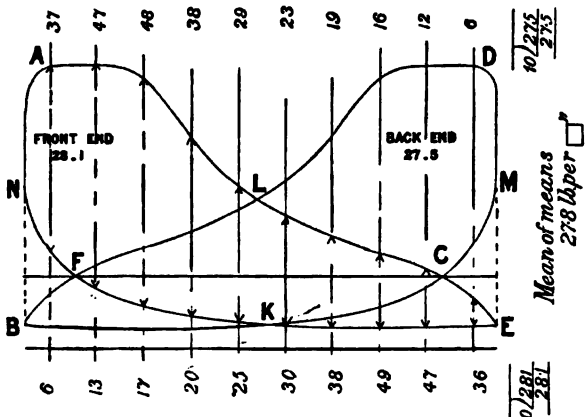


FIG. 22.—Calculation of Méan Pressure.

ten vertical lines (as shown in Fig. 22), the two outer parts forming one part, each being one-half. The heights contained between the upper and lower parts of each figure are measured for each stroke separately, and the mean height is calculated. The mean of the two means so obtained represents the mean pressure on the piston during one revolution—two strokes.

For convenience in measuring diagrams, a planimeter is used when a large number of diagrams are to be calculated; but for ordinary practice long slips of paper are run over the diagrams, and the heights for each stroke are ticked off in turn, from point to point, along one edge. This saves the labour of adding, and is less liable to error than taking each height separately. The total of these heights, divided by ten (the number of ordinates), gives the mean

height and mean pressure to the scale on which the diagram is taken.

Indicated Horse-Power of an Engine.—A general idea of the horse-power of an engine is obtained by considering the resistance overcome by the steam when acting on the piston. A force equal to the difference in pressure (*i.e.* the effective pressure) on the two sides of the piston is used to overcome that resistance. The piston of an ordinary reciprocating engine moves with a varying velocity when rotating a crank, but if the crank makes N revolutions in each minute, the piston then travels through 2N strokes, or twice the number of revolutions per minute. The force, therefore, acts through a distance, or creates motion by overcoming resistance through the same distance, per minute of 2NL feet, when L is the length of each stroke in feet.

The work done per minute is thus = F × 2NL foot-lb.

Now, as we have seen—

$$F = \left\{\begin{matrix}\text{the mean effective pressure per} \\ \text{square inch on the piston}\end{matrix}\right\} \times \left\{\begin{matrix}\text{area of the piston} \\ \text{in square inches}\end{matrix}\right\} = P \times A$$

and work done per minute = P × A × 2L × N in foot-lb. per minute.

Also 1 H.P. = 33,000 foot-lb. per minute;

$$\text{therefore I.H.P. of engine} = \frac{2PLAN}{33{,}000}.$$

The mean effective pressure on the piston is generally found in practice by using an indicator, and the power thus calculated by its use is called the Indicated Horse-Power (I.H.P.). The I.H.P. developed in each cylinder of an engine is calculated separately, so that the total I.H.P. of the engine is the sum of the powers obtained in the several cylinders.

Calculation of Cylinder Constant.—As an example, take the cylinder diameter at 27 inches, piston-rod diameter at $5\frac{1}{4}$ inches, and length of stroke at 33 inches. Find the cylinder constant.

The effective area on the forward stroke in the direct-acting engine is equal to the area of the piston, if there is no tail-rod; on the return stroke the effective area is reduced by the area of the piston rod. For absolute accuracy the constants for the forward and return strokes should be measured separately, and multiplied by their respective mean pressures, and the mean taken

of the two results thus obtained. In practice such extreme accuracy is unnecessary, and the mean of the two areas is taken as the value of A; and the mean of the two mean pressures is taken as the value of P (in formula for calculating the I.H.P.).

In the example, therefore—

$$\begin{aligned} A &= \tfrac{1}{2}\{27^2 \times 0{\cdot}7854 + 27^2 \times 0{\cdot}7854 - (5\tfrac{1}{4})^2 \times 0{\cdot}7854\} \\ &= \tfrac{1}{2}\{572{\cdot}557 + 572{\cdot}557 - 21{\cdot}648\} \\ &= \tfrac{1}{2} \times 1123{\cdot}466 \\ &= 561{\cdot}733 \text{ square inches} \end{aligned}$$

$$\text{The cylinder constant} = \frac{2 \,.\, \text{L} \,.\, \text{A}}{33{,}000} \text{ (and L} = \tfrac{33}{12}\text{)}$$

$$= \frac{33}{12} \times \frac{1123{\cdot}466}{33{,}000}$$

$$= 0{\cdot}094622$$

Revolution Constant.—For each engine, where diagrams are frequently taken, a table is generally compiled of revolution constants. Thus, this constant for the above example, at 110 revolutions, is—

$$0{\cdot}094622 \times 110 = 10{\cdot}40842.$$

It is now only necessary to multiply this constant by the mean pressure obtained at the corresponding number of revolutions to obtain the I.H.P.

For each cylinder of a stage-expansion engine a cylinder constant exists, and if the corresponding revolution constant be multiplied by the mean pressure indicated by the diagram, the product is equal to the power indicated in the cylinder. The sum of the powers thus obtained separately for each cylinder is the total horse-power indicated by the engine, or total I.H.P.

Brake Horse-Power.—The indicated horse-power is a measure of the work done per minute on the piston. Some of this power is used in overcoming the resistance of the engine itself to motion, such as frictional resistances of the piston in the cylinder, of the piston-rod packing, and of the bearings, so that only a certain proportion is actually utilised in rotating the shaft. This proportion is called the brake horse-power, and for small motors, whose energy can be entirely utilised for the test, is generally measured by means of a brake which absorbs the rotational energy of

the shaft. (This form of brake is also called an *Absorption dynamometer.*)

In Fig. 23, *S* is a pulley on the shaft which is rotated by the engine. Two nearly semicircular blocks of wood, *C*, *C*, are bolted together as shown about the pulley. To the upper block is secured a long lever *D*, and a short lever *E*, and at *E* a weight is placed so that *E* and *D*, and the apparatus generally, balance about the shaft centre, the bolts being then sufficiently slacked to allow free motion when the shaft is not in motion. A spring balance, or adjustable weight, can be suspended to the outer end *D* of the long lever.

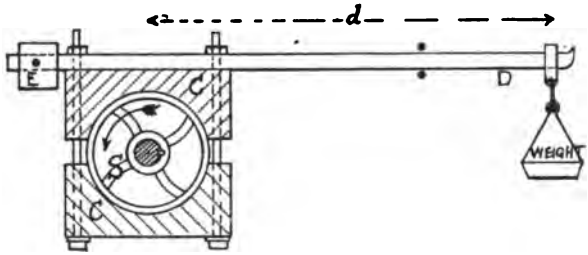


FIG. 23.—Prony Absorption Dynamometer.

Let R = the radius = $\frac{1}{2}$ diameter of the pulley;

F = frictional resistance of pulley in the brake blocks, which must be tightened up as necessary.

Equating the moments of the forces acting about the centre of the shaft, the lever remains horizontal when $FR = Wd$, or when $F = \frac{Wd}{R}$.

The work done against the resistance (F) in one revolution of the pulley

$$= F \cdot 2\pi \cdot R \text{ foot-lb.};$$

and if the shaft turns N times per minute—

$$\text{the brake H.P.} = \frac{F \cdot 2\pi \cdot R \cdot N}{33{,}000} = \frac{W \cdot 2\pi \cdot d \cdot N}{33{,}000}.$$

In practice the engine is designed to run at a certain number of revolutions per minute, and is actually run at that speed when finding the B.H.P.

EXAMPLE.—In H.M.S. *Drake* a pull of 3000 lb. at a radius of 8 feet 9 inches is required to move the after-length of propeller shafting (friction test

in dry dock); what H.P. is necessary to overcome the stern shaft friction when the shaft makes 120 revolutions per minute (full speed)?

$$\text{H.P.} = \frac{W \times 2\pi \times d \times N}{33{,}000}$$

$$= \frac{3000 \times 2\pi \times 8\frac{3}{4} \times 120}{33{,}000}$$

$$= 600, \text{ about, for each of the twin shafts.}$$

When developing the full power of 30,000 I.H.P., about 1200 H.P. is thus absorbed, amounting to about 4 per cent of the full power. For a new ship the percentage would be much less; but taking into consideration that the stern glands were temporarily unpacked, and special precautions taken to reduce friction by running water through the stern bearings, and that friction is also overcome in several plummer blocks not included in the test, the result is probably fairly accurate for working conditions.

Absorption dynamometers are only suitable for measuring low and moderate powers. They are commonly used for testing the H.P. (actually Brake H.P.) of engines used for driving motor cars, and the two figures usually quoted in the trade catalogues correspond to the B.H.P. of the axle shaft at two differing speeds which may be obtained under heavy and light loads. The B.H.P. can also be obtained by measuring the difference in tension of a belt or of a spring used in transmitting the energy of one shaft to another. A simple and fairly accurate arrangement of *belt transmission dynamometer* is shown in Fig. 24 (Thornycroft).

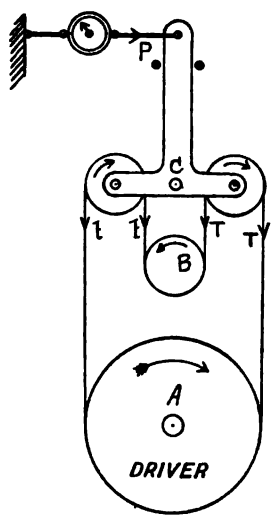


FIG. 24.—Thornycroft Transmission (belt) Dynamometer.

The belt is led over two loose pulleys, as shown, from the driver A to the driven shaft B. The loose pulleys are held by an inverted T-frame pivoted at C. The upper end of it is connected with a spring balance, P, or other simple device. When the shaft is revolved at its proper number of revolutions, neglecting friction of the pulleys, the pulls in the belt are respectively T and t; and the work transmitted per revolution $= (T - t) \times 2\pi \times R$ in foot-lb., where R = radius of pulley B in feet.

Equating the turning-moments about C, and supposing the loose pulleys to be equidistant L from C, which is usually arranged—

$$2 \cdot T \cdot L = 2 \cdot t \cdot L + P \cdot CP$$

$$T - t = \tfrac{1}{2} P \cdot \frac{CP}{L}.$$

Knowing the lengths CP and L and the radius R, and observing the pull P—

$$\left.\begin{array}{l}\text{The } \textit{brake horse-power} \text{ transmitted} \\ \text{at N revolutions per minute}\end{array}\right\} = \frac{P \cdot \pi \cdot R \cdot N \cdot CP}{33{,}000 \times L}.$$

For engines driving dynamos, the *electrical* horse-power can be obtained by observing the ammeter and voltmeter; then—

$$\text{Electrical H.P.} = \frac{\text{ampères} \times \text{volts}}{746},$$

which is generally 90 to 95 per cent of the B.H.P.

$$\begin{aligned}\text{Electrical H.P.} &= \tfrac{9}{10} \times \text{B.H.P.} \\ &= \tfrac{9}{10} \times \tfrac{3}{4} \times \text{I.H.P.} \\ &= \tfrac{27}{40} \times \text{I.H.P.}\end{aligned}$$

For rough general purposes, the I.H.P. of an engine driving a dynamo—

$$\text{I.H.P.} = \frac{\text{ampères} \times \text{volts}}{500}.$$

But this does not apply to turbines whose I.H.P. cannot be measured and whose mechanical efficiency is greater.

Shaft Horse-Power.—For the measurement of power there are many methods, all more or less elaborate. Generally speaking, fairly accurate results are obtainable with an indicator suitably adapted to the special type of engine under test. It is quite out of the question to assume that any particular indicator is suitable to all cases, and especially is this true of the particular case of marine motor-engines running at high rates of revolution and with great range of pressure within the cylinder. With absorption dynamometers of the Prony brake type the whole of the shaft-power is absorbed by the brake, and therefore they are useless for determining the power transmitted from an engine to some useful purpose.

The measurement of the shaft-power affords the most valuable information, both for the engineer and the naval architect, because all losses due to friction of the mechanism are eliminated. When transmitting power the shaft naturally withstands a stress, compounded of the actual twist imparted to it and the thrust-compression produced by the action of the propeller, which thrust in ordinary cases may, according to Mr. Hamilton Gibson's experiments, amount to as much as 18 per cent of that due to torque alone. The torque, it need hardly be mentioned, is the turning-moment in, say, ft.-lb. required to produce a certain angular twist. The torsion-meter is used to measure this angular twist between two points of a shaft, and from this angle the shaft horse-power is calculated. For accurate measurement the shaft between these two points must have the same external diameter throughout, and, if hollow, the same internal diameter, and also the material should be homogeneous. These matters are generally in accord with good practice; the shaft is machined externally and internally, and the material is generally of the best manufacture. If a coupling be included between the two points of measurement, it is generally found that, considering that no twist is imparted in its length, by deducting such length from the actual distance between the two points of measurement a sufficiently accurate correction is made. In ordinary practice, therefore, it is only necessary to know the modulus of elasticity applicable to the material, and to apply such corrections as may appear necessary, such as the one mentioned above.

For particular cases, with material of unknown elastic quality, a method applied by Mr. Gibson during his experiments may be interesting. He applied a torque on a shaft under consideration by means of two spring-balances at opposite ends of a couple which tended to eliminate friction in the supporting bearing. At the initial, or dead point of measurement, the shaft was fixed and prevented from rotation. At the same time the shaft was compressed, by means of a bell-crank lever, to represent the propeller-thrust, which was assumed to be about 65 per cent of that due to the shaft horse-power. A ball-bearing was introduced between the end of the shaft and the compression lever to enable the shaft to turn freely. During an experiment of this kind it is necessary to keep the shaft "alive" by tapping its surface with lead hammers, because there may be some tendency to "lag" when putting the

load on and off. Direct mechanical measurement appears most favourable to perfect accuracy, and it is believed that in the Chadburn Flash-light Torsion-meter (Bevis and Gibson's patent) all backlash elements are practically eliminated. This apparatus depends for its accuracy on the almost infinite velocity of light, the absolute straightness of a ray of light, and the automatic instantaneous shutter action of two opposite slots in two discs fitted rigidly on the shaft at a reasonable distance apart.

Fig. 25 shows an outline profile of a shaft and apparatus with three plans under various conditions, and the general construction of the main features of the apparatus for use with turbine engines and reciprocating engines respectively. Each apparatus consists of two metal discs, A and B, rigidly secured to the shaft, and for *turbine engines* each disc has a single small radial slot, C, near the periphery, accurately adjusted so as to lie exactly in the same radial plane containing the axis of the shaft. The discs and edges of the slots are painted a dead black so as to reflect as little as possible, and thus not confuse the single beam from the lamp. The portable lamp, D, is accurately secured behind one disc, A, so that the light shows through both discs when the slots are in opposition. The "finder," F, is also portable, but so made as to be readily mounted behind the other disc, B, and secured quickly and accurately in position. A movable eye-piece, E, is fitted to the finder F, and E is adjustable circumferentially by means of a delicate micrometer screw gear which sets the finder with extreme accuracy. There is a machine-divided scale, G, and a vernier, H, on the finder, which is also provided with a magnifying reader, J, to enable the scale to be read off to $\frac{1}{100}$th of a degree, or $\frac{1}{36000}$th of the circumference of the circle described by the slots when the discs revolve. The torque-finder is the most delicate part of the apparatus, and when not in use should be unshipped and carefully stowed in its box. The discs should also be suitably guarded by casings if left in place.

Before taking an actual reading, it is necessary to correct the accuracy of the apparatus by taking a reading with the shaft revolving under no load. This may be done by disconnecting the propeller, or by allowing the shaft to be dragged round for a few moments by the "way" on the vessel. This correction of the "zero" reading must, of course, be applied to any subsequent observations when transmitting a load.

When transmitting a load the torque-finder is moved until

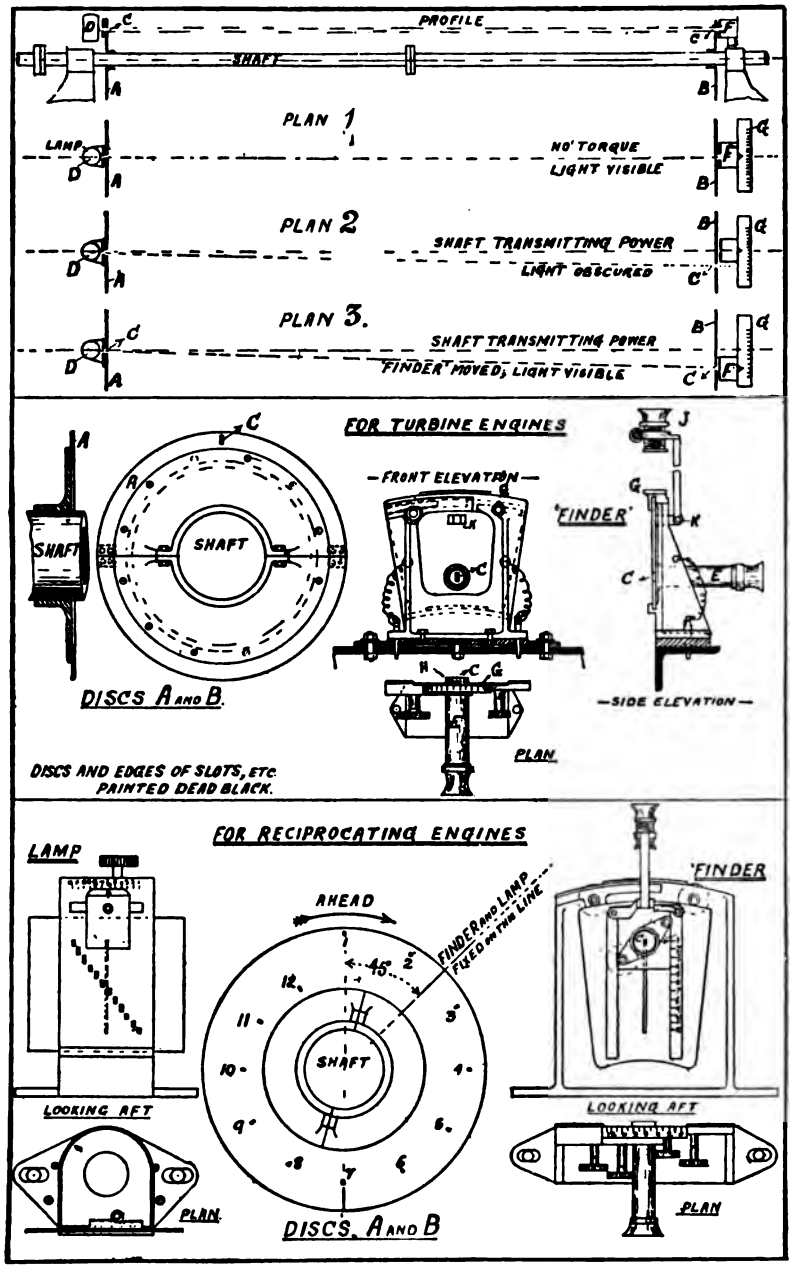


FIG. 25.—Flash-light Torsion-Meter.

the exact angle of the twist of the shaft allows the flash to be clearly seen through the eye-piece. For a rapidly revolving shaft, such as the propeller-shaft of a turbine steamer, the light as seen through the torque-finder is practically continuous; at lower speeds the light appears to flicker, but there is no difference either in the accuracy of the reading or in the ease with which the torque is obtained. Typical of this accuracy may be cited the following experience with a Channel turbine steamer:—Running at full power, about 2600 horse-power on one of three shafts, and at 470 revolutions per minute, torsion-meter records were taken, as well as at various progressive speeds down to 24·4 R.P.M., when only 9·2 shaft horse-power was developed. During the trials of a 33-knot destroyer running at about 750 revolutions per minute, one observer was able to get the horse-power of the three separate shafts while running each measured mile, *i.e.* in less than two minutes.

In cases where a long length of shaft is not available, a modification in the disposition of the apparatus is made, and the flash is thrown out radial to the shaft instead of in the direction of its axis. This arrangement involves more material in the apparatus, and it is somewhat more difficult to fit up; but the accuracy is maintained, and readings can be well obtained from a piece of shaft not more than two or three diameters long.

For the shafting of *reciprocating engines*, or any shaft exposed to a fluctuating torque, means are provided to enable the observer to take twelve or more separate readings for one revolution. For this purpose the light and the eye-piece are moved radially to or from the shaft so as to come opposite each pair of slots in the discs, and observations made for each position. When these torque readings are plotted out on a development of the crank circle, as in a twisting-moment diagram, the resultant curve is found to agree very closely with the diagram built up from the corresponding set of indicator-diagrams taken from the engine cylinders. The difference between the two curves shows, with remarkable fidelity, the power absorbed by engine friction (see Fig. 272).

The calculation of the shaft horse-power is based on Rankine's formula, and for this purpose it is necessary to know or to observe:—

A = angle of twist of shaft, in degrees;
N = number of revolutions per minute of the shaft;

D = external diameter of the shaft, in inches;
d = internal diameter of the shaft, in inches;
(when the shaft is solid, $d = 0$)
L = length of shaft between the points of measurement, in inches;
(if a coupling is included its length should be deducted)
K = constant dependent on the modulus of elasticity;
= 3·27 for a steel shaft with a modulus of 11,250,000.

Then—

$$\text{Shaft H.P.} = \frac{\text{A.N.}\ (D^4 - d^4)}{\text{K.L.}}.$$

For the same shaft and measuring apparatus, the values of D, d, K and L, do not change, so that—

$$\text{Shaft H.P.} = \text{C.A.N.}$$

where C = a constant for the particular shaft

$$= \frac{D^4 - d^4}{\text{K.L.}}$$

= shaft constant.

The calculation of the shaft horse-power thus resolves itself into a simple sum in multiplication; and for each ship, by tabulating the values of C.N. (the revolution constant) the calculation can be still further simplified.

Efficiency.—In every mechanism or engine there is a certain wasteful expenditure of energy, so that the useful work done is always less than the energy expended. The efficiency of a machine is represented by a ratio, fraction, or percentage—

$$\text{Efficiency} = \frac{\text{useful work done (in ft.-lb.)}}{\text{energy expended (in ft.-lb.)}}.$$

In marine propulsion there are *five* elements, each of which absorbs a certain proportion of the energy expended. These elements are: the *boiler*; the working substance, *steam*; the engine *mechanism*; the *thrust* mechanism; and the *propeller*. Finally, only a very small proportion of the energy of the fuel is expended in usefully urging the ship along. A few typical examples, estimated

on such details as are available, show the *total* or *overall efficiencies* of several modern marine propelling plant, working *at full power.* The overall efficiency is the product of the efficiencies of the five elements, and, in the table below, each efficiency is shown as a percentage.

	Boiler.	Steam.	Engine.	Thrust.	Propeller.	Overall.
H.M.S. *Topaze* (reciprocating)	53 ×	11 ×	90 ×	85 ×	65 =	3 per cent.
H.M.S. *Amethyst* (turbines)	52 ×	16 ×	95 ×	95 ×	55 =	4¼ ,,
H.M.S. *Drake* (reciprocating)	70 ×	13 ×	90 ×	90 ×	68 =	5 ,,
H.M.S. *Dreadnought* (turbines)	70 ×	17 ×	98 ×	98 ×	58 =	6¾ ,,
ss. *Caronia* (reciprocating)	75 ×	16 ×	90 ×	90 ×	69 =	6¾ ,,
ss. *Carmania* (turbines)	75 ×	18 ×	98 ×	98 ×	60 =	7¾ ,,

At lower powers generally, the efficiencies of the boiler and propeller increase for both reciprocating and turbine propelled vessels; the efficiency of the steam generally increases for reciprocating and decreases for turbine engines; the efficiencies of the engine and thrust decrease for both reciprocating and turbine engines. The various efficiencies are explained briefly below, and further information is given in fuller detail later on in the book.

1. *The Boiler.*—The heat energy contained in the fuel consumed is not all transferred to the steam generated in the boiler. There are losses from imperfect combustion; from heat passing away by radiation, conduction, and other causes, to the stokehold and its surroundings; and by the passage of the hot gases through the funnel into the atmosphere. Finally, instead of 15 lb. of water being converted into steam, from and at 212° F., by burning 1 lb. of coal, only 10 to 12 lb. weight of steam leave the boiler, and thus—

$$\text{Efficiency of boiler} = \frac{10 \text{ to } 12}{15} = 0{\cdot}66 \text{ to } 0{\cdot}8 = 66 \text{ to } 80 \text{ per cent}$$

When the fires are urged by excessive forced-draught, the efficiency is low, as shown above by the *Topaze* and *Amethyst*; at lower powers the efficiency becomes 65 for *Amethyst* and 62 per cent for *Topaze.* This comparatively low efficiency is attributable to the closed stokehold system and the absence of heat-saving appliances, such as hot-air draught and feed heaters or

economisers. The *Drake* is fitted with economisers, and both the *Drake* and *Dreadnought* boilers are forced to only a moderate extent. The *Caronia* and *Carmania* are fitted with Howden's closed ashpits and a system of heating the air, by the waste funnel gases, before it is supplied to the fires.

2. *The Steam.*—Of the total heat energy leaving the boiler in the form of steam there is a moderate loss by radiation, etc., and the exhaust enters the condenser principally as steam, but partly as water. The heat in the exhaust steam and water, commonly known as the *exhaust waste*, or "heat rejected," is absorbed by the condenser circulating-water and pumped into the sea. To a very large extent the exhaust waste is unavoidable, therefore—

Efficiency of the steam $= \frac{1}{10}$ to $\frac{1}{5} = 0{\cdot}1$ to $0{\cdot}2 = 10$ to 20 per cent.

Turbine engines give a greater efficiency than reciprocating engines at full power; but at powers below about one-half, the reciprocating engine is more efficient, unless cruising turbines are fitted, when the power may be reduced to about one-sixth the full power with about an equal efficiency.

3. *The Mechanism of the Engine.*—Where the brake horse-power or shaft horse-power is measured at the part of the shaft between the engine coupling and the thrust-block, the

$$\text{Efficiency of the mechanism} = \frac{\text{brake horse-power}}{\text{indicated horse-power}}.$$

Of the work done on the pistons (actually measured by the use of the indicator and known as indicated work), some is absorbed in overcoming frictional resistances of the slide valves, pistons, packings, guides, bearings and other working parts, and thus only about 75 per cent at low powers, up to 94 per cent at full power, is transferred into useful work in rotating the shaft—

Efficiency of the mechanism $= 0{\cdot}75$ to $0{\cdot}94 = 75$ to 94 per cent.

The frictional resistances of the engines of H.M.S. *Hyacinth* and *Minerva*, without any load (propellers removed, and therefore also without thrust resistance) and at full speed, were found to absorb 5 per cent of the full I.H.P. Working under a load the frictional resistances must necessarily be greater, because there is an

increased pressure on all bearing surfaces, packing, etc., together with an increased temperature of all internal working parts.

By using a Fottinger flash-light apparatus for measuring the shaft H.P. of the *Kaiser Wilhelm II.*, and the German cruiser *Hamburg*, it was found that 7 and 6 per cent of the I.H.P. respectively was absorbed in engine friction at full power; but at half-power the mechanical efficiency was reduced to about 82 or 80 per cent. Roughly, at low powers of reciprocating engines the mechanical efficiency

$$= E\sqrt{\frac{n}{N}},$$

where E = efficiency at full power;
N = revs. per minute at full power;
n = revs. per minute at the lower power.

With turbine engines the internal friction is less than with reciprocating engines, and the mechanical efficiency is seldom less than 95 per cent, and at full power with recently constructed turbines it may be taken as 98 per cent.

4. *The Thrust Mechanism.*—There is very little exact evidence on this important element of ship propulsion, and consequently it is very difficult to form an estimate. From experiment it is known that within the limits of moderate powers, corresponding to speeds between 10 and 22 knots in a 24-knot ship, for instance, the I.H.P. increases as the *cube* of the speed; but friction in the thrust mechanism, increasing with the pressure on the thrust collars in urging the ship along, probably increases as about the *square* of the speed. Thus the I.H.P. absorbed in overcoming the frictional resistance of the thrust

$$\propto S^2,$$

where S = speed of the ship in knots.

If the ship be towing another, or be prevented in some way from moving or attaining her normal speed through the water, the thrust pressure and friction increase very rapidly, so that in the above formula the speed of the vessel must be considered and not the actual revolutions of the propeller. Also, from the above formula it is evident that the efficiency of the thrust mechanism decreases for powers below the full power.

From some experiments, thrust friction absorbs 6 or 7 per cent of the full I.H.P. at full speed; but in ordinary cases with the best form of lubrication (see Thrust-Block), it is probably at least 10 per cent, and in the older ships with indifferent lubrication it is probably nearer 20 per cent.

With Parsons' turbines, at present the only notable exception, nearly the whole of the propeller thrust is absorbed by the statical difference of steam pressure within the turbine itself, without any expenditure of energy, and only a small fraction, probably about 2 per cent of full power for recently constructed turbines, is absorbed by the thrust mechanism.

At full power, for reciprocating engines—

Efficiency of thrust mechanism = 80 to 90 per cent,
and for Parsons' turbines = 95 to 98 per cent.

5. *The Propeller.*—Energy is lost by screw propellers from the oblique action of the blades, only a moderate proportion of the water acted on being moved in a direction opposite to that of the ship, and the remainder is driven away in other directions which do not assist the ship along; other disturbances, such as eddies, are also caused in the water passing between the blades. There is considerable friction between the water and the blade surfaces, which tends to increase disturbance and wasteful effect of oblique action; and there is also cavitation in some cases.

Screw propellers driven by turbines are generally rotated at a greater number of revolutions per minute than those driven by reciprocating engines (about twice as great in the *Amethyst*), and, consequently, there is greater frictional resistance to the passage of the screw blades through the water, and there are correspondingly greater wasteful disturbances.

At full power, with reciprocating engines—

Efficiency of propeller = 65 to 70 per cent,
and with turbine engines = 55 to 60 per cent.

The proportion of the I.H.P. which is effectively used in propelling the ship (that is, the *product* of the efficiencies of the engine *mechanism*, *thrust*, and *propeller*, and known as the *effective* or *propulsive* H.P.) is seldom greater than 55 per cent for reciprocating engines, and is probably slightly less for turbine

engines, although this is a matter of some doubt owing to the difficulty of estimating the I.H.P. of turbines.

Maximum Efficiency of a Perfect Heat Engine.—In a heat engine using a perfect gas as the working substance, the quantity of heat, Q, contained in unit mass of the gas is directly proportional to its absolute temperature, T_0. Only a certain portion of the total quantity of heat in the substance can be changed into useful work, because there is a certain absolute temperature, t_0, below which the engine cannot be worked. The mass of gas, therefore, leaves the engine with a quantity, q, of heat in it, which corresponds evidently to an absolute temperature, t_0.

The heat which can be utilised $= Q - q$, and the efficiency then obtained becomes—

$$= \frac{\text{heat usefully expended}}{\text{total heat available}} = \frac{Q-q}{Q} = 1 - \frac{q}{Q},$$

in which $\frac{q}{Q} = \frac{t_0}{T_0}$.

Therefore efficiency $= 1 - \frac{t_0}{T_0} = \frac{T_0 - t_0}{T_0}$, in *absolute* temperatures;

or $$= \frac{T-t}{T+461},$$

where T and t are measured from the ordinary zero, 0° Fahr.

This ratio $\frac{T-t}{T+461}$ is termed the maximum efficiency of a perfect heat engine, and is frequently used for comparison of heat engines, one with another. The ratio or formula for maximum efficiency, although applicable to the steam engine as a heat engine, is by no means absolutely true. The actual quantity of heat contained in a mass of steam, either on entry to or exhaust from the steam engine, is dependent on the dryness fractions of the steam, both of which may vary considerably, although the temperatures, T and t, may remain constant. This variation from actual truth does not, however, detract from its usefulness in the general comparison of heat engines, whether using the same or some different working substance, and the following tabular statement of the maximum possible steam (heat) efficiencies under various conditions is instructive:—

		Possible increase per cent.
For Newcomen's engine	$= \frac{212-150}{212+461} = \frac{62}{673} = 0{\cdot}092$ about.	
For a non-condensing engine, 50 lb. pressure	$= \frac{297-212}{297+461} = \frac{85}{758} = 0{\cdot}112$ about.	22
For a condensing engine, 50 lb. pressure	$= \frac{297-100}{297+461} = \frac{197}{758} = 0{\cdot}26$ nearly.	132
For a condensing engine, 100 lb. pressure	$= \frac{338-100}{338+461} = \frac{238}{799} = 0{\cdot}298$ nearly.	14·6
For a condensing engine, 150 lb. pressure	$= \frac{366-100}{366+461} = \frac{266}{827} = 0{\cdot}321$ about.	8
For a condensing engine, 200 lb. pressure	$= \frac{388-100}{388+461} = \frac{288}{849} = 0{\cdot}34$ nearly.	$6\frac{1}{4}$
For a condensing engine, 250 lb. pressure	$= \frac{406-100}{406+461} = \frac{306}{867} = 0{\cdot}353$ about.	3·8
For a condensing engine, 300 lb. pressure	$= \frac{422-100}{422+461} = \frac{322}{883} = 0{\cdot}364$ about.	3·1

SUMMARY OF FORMULÆ

966 = Latent heat of 1 lb. saturated steam under atmospheric pressure and a temperature of 212° F.

$L = 966 - 0{\cdot}7\ (T - 212)$

$S = T - t$

$H = L + S$

$= 966 - 0{\cdot}7\ (T - 212) + (T - t)$, for saturated steam

$= x\,.\,L + S$, for wet steam

$= x\,.\,\{966 - 0{\cdot}7\ (T - 212)\} + (T - t)$

1 British Thermal unit = 780 ft.-lb. = J = Joule's equivalent.

Expansion curves—

Hyperbola, $P\,.\,V = \text{constant}$.

Saturation, $P\,.\,V^{1{\cdot}0645} = \text{constant}$.

Adiabatic, $P\,.\,V^{1{\cdot}035+\frac{x}{10}} = \text{constant}$.

Equivalent evaporation, from and at 212° F.—

$$= \frac{n.(x\,L + S)}{966}.$$

Maximum efficiency of a heat engine using a perfect gas as the working substance—

$$= \frac{T - t}{T + 461}$$

$$\text{I.H.P.} = \frac{2.P.L.A.N}{33{,}000}$$

$$\text{Brake H.P.} = \frac{W.2\pi.d.N}{33{,}000}$$

$$\text{Shaft H.P.} = \frac{A.N.D^4}{K.L} \text{ for a solid shaft}$$

$$= \frac{A.N.(D^4 - d^4)}{K.L} \text{ for a hollow shaft.}$$

PART II

CHAPTER VI

MARINE BOILERS—WATER-TANK TYPE

THE marine boiler of the present day is essentially a high-pressure boiler, and to a large extent it depends for its power of converting water into steam on the area of tube surface which it contains. The boiler is called a *tank* boiler if the gases pass through the tubes on their way to the uptake and funnel, or chimney; this boiler is usually *internally* fired, and the furnaces and the tubes are surrounded by water. If the gases pass around and between the tubes, the boiler is *externally* fired, and the tubes contain water and sometimes partly steam; the boiler is then of the *water-tube* type.

In this chapter the boilers described are of the tank type, and consist principally of cylindrical boilers with nearly flat ends. They are as follows:—

Return tube cylindrical boiler, single ended.
„ „ „ double ended.
Direct tube cylindrical boiler, now seldom used; and
„ „ locomotive boiler, as used on railways, etc., but abandoned for marine use in favour of upright water-tube boilers.

The various types of water-tube boilers may be divided into two fairly well-defined classes, namely, those in which the—

1. Tubes are nearly *upright*, and in consequence of which the natural circulation is very rapid, and the tubes are of small diameter. (Chap. VII.)

2. Tubes are nearly *horizontal*, and in consequence of which the natural circulation is comparatively slow, and the tubes are

necessarily of large diameter to prevent overheating of the material. (Chap. VIII.)

At the end of Chap. VIII. there are some notes on the "Influence of Practice on Boiler Construction and Design" which are of a somewhat more advanced character than the descriptions of the various types of boilers. The notes, however, shed considerable light on the direction of future construction.

Historical Introduction.—Before and during the early Newcomen period the ordinary boiler was naturally something of the shape of a kettle or covered saucepan. Newcomen adopted an almost globular shape, and this continued to be used for many years. It must be remembered that in those days iron or steel plates of uniform thickness were almost unknown, and that the plating of the early boilers had to be hand forged from the pig-iron blocks. Some of the smaller boilers were constructed of two dished plates as shown in Fig. 4.

With the demand for greater powers the boiler itself became a larger affair, and a boiler, very similar in shape to a *haystack*, but built of many plates riveted together, became a common form. At the beginning of the nineteenth century the boiler developed into a cylindrical shape with rounded ends, but a more general type was *box* shaped; both these boilers were *internally* fired, that is, flues or furnaces were introduced into the interior of the boiler, and, generally, each flue was surrounded by water. The "*Galloway*" *boiler* of the present day is a survival, in an improved form, of the early cylindrical round-ended boiler. The pressure in the early boilers seldom exceeded 5 or 10 lb. per square inch, although Watt, who clearly understood the advantage of higher pressures, attempted to obtain a boiler for 50 or 60 lb. working pressure.

No great advance was made on the *flue boiler* until about 1840, when the *Penelope* frigate was fitted with boilers of box shape, containing a large number of tubes, and called *multitubular boilers.* With the exception of improvements in material and design the multitubular box boiler was in general use in the Royal Navy until 1870; but these improvements allowed an increase in pressure from 5 or 10 lb. up to 35 or 40 lb. per square inch. In tubular boilers, the gases and flames from the furnaces pass through the tubes on their way to the uptake and funnel, and the tubes are generally horizontal and surrounded by water.

Cochran's boiler took the form of a mixed flue and water-tube boiler; the tubes were vertical and filled with water, and the gases

surrounded the tubes on their passage to the uptake. This boiler gave an increased efficiency for a short period; but the smoke passages could not be properly swept while under steam (a defect shared by many water-tube boilers more or less well known at the present day), and the efficiency fell in a few hours to something below that of the ordinary multitubular type.

From the multitubular box-boiler grew the *cylindrical marine boiler* of the present time. Some boilers (called *oval*) were flattened at the sides with a semicircular top and bottom. The pressure rose in some instances to 60 lb. per square inch with the introduction, about 1860, of the surface condenser, which contributed very largely to the possibility of using increased pressures by decreasing the frequency of the necessary scaling and cleaning. With higher pressures came the introduction of compound or stage-expansion engines, and the *Alexandra* (1871) was one of the first naval vessels to be fitted with cylindrical boilers and compound engines working at 60 lb. pressure. A slight increase of pressure followed, but was adopted slowly, until the introduction about 1880 of the evaporator solved the difficulty of scale forming in the boilers. From this time the growth in pressure was rapid, and in 1900 was commonly 300 lb. for naval boilers of the water-tube type, and 200 to 210 lb. for cylindrical boilers. Beyond these pressures there has been no general advance except in a few particular instances.

The general introduction of water-tube boilers took place about 1893-94, and was brought about by a desire to obtain a greater power with safety on less weight—an object which has been very distinctly attained in all modern naval vessels, the total weight of boilers and machinery in which is at most one-half that of mercantile vessels. The marine cylindrical boiler will produce continuously and safely from 15 to 18 I.H.P. per square foot of grate surface, equivalent to a coal consumption of $22\frac{1}{2}$ to 27 lb. of coal per square foot of grate. Any attempt to increase the consumption beyond this amount is likely to produce permanent defects in the boiler. The limit is very clearly defined for boilers of the usual cylindrical construction. With naval water-tube boilers a far greater consumption can be safely obtained, varying from 30 to 100 lb. per square foot of grate area, corresponding to the type of boiler and the angle of inclination of the water-tubes. If the tubes are nearly horizontal the circulation is naturally slow, and to prevent overheating the diameter of tube and water space must be large; but if

the tubes are upright, the diameter of tube and water space may be small; and herein lies the essential difference between so-called *large-tube* and *small-tube* water-tube boilers. Now consider a space 1 square yard in area and 8 feet long, and it is evident that a much greater number of small tubes than large tubes can be fitted into it, allowing the same distance between any two adjacent tubes; in other words, a much larger boiler heating surface can be fitted into an equal space and volume; and in destroyers, and recently in the largest vessels, this has been done with a saving in weight as well as volumetric space. In water-tube boilers the brickwork and casings are of considerable importance, and on their continued efficiency depends the efficiency of the boiler, other things being equal. In a *large-tube* boiler with nearly horizontal tubes, such as the Belleville, Babcock, Niclausse, and Dürr boilers, the brickwork and casings around the furnaces are exposed to a temperature of 2500 to 3000 degrees F. on four sides; in the upright tube type, such as the White-Forster and Yarrow, this high temperature only touches the front and back. As the various boilers are described in this and other chapters, the differences become apparent; and, without prejudice, each boiler should be rated on its own merits.

The following table (*Engineering* of August 2, 1907) shows the effect of the rise of boiler pressure on the coal consumption, power, and speed, etc. of a few typical Cunard Atlantic Liners:—

	1840. *Britannia.*	1856. *Persia.*	1879. *Gallia.*	1884. *Umbria.*	1893. *Campania.*	1907. *Lusitania.*
Steam pressure (lb.) . .	9	33	75	110	165	200
Coal necessary to steam to New York (tons)	570	1400	836	1900	2900	5000[1]
Coal per I.H.P. per hour (lb.)	5·1	3·8	1·9	1·9	1·6	1·45[1]
I.H.P. per ton of coal per hour	440	590	1180	1180	1400	1545[1]
Indicated horse-power . .	710	3600	5000	14,500	30,000	68,000
Speed (knots) . . .	8·5	13·1	15·5	19	22	25[1]
Displacement (tons) . .	...	...	...	10,500	18,000	38,000

[1] Estimated—no actual details published.

Return-tube Cylindrical Boiler.—A circular shell pressed outwards by an internal pressure uniformly distributed over its inner surface, as in a boiler, tends to maintain its shape. This principle is embodied in the cylindrical boiler; but the flat front and back tend to bulge out, like a sail under the pressure of the wind,

and to prevent this happening stays are fitted from front to back. These, as shown in Fig. 26, however, all run parallel to each other,

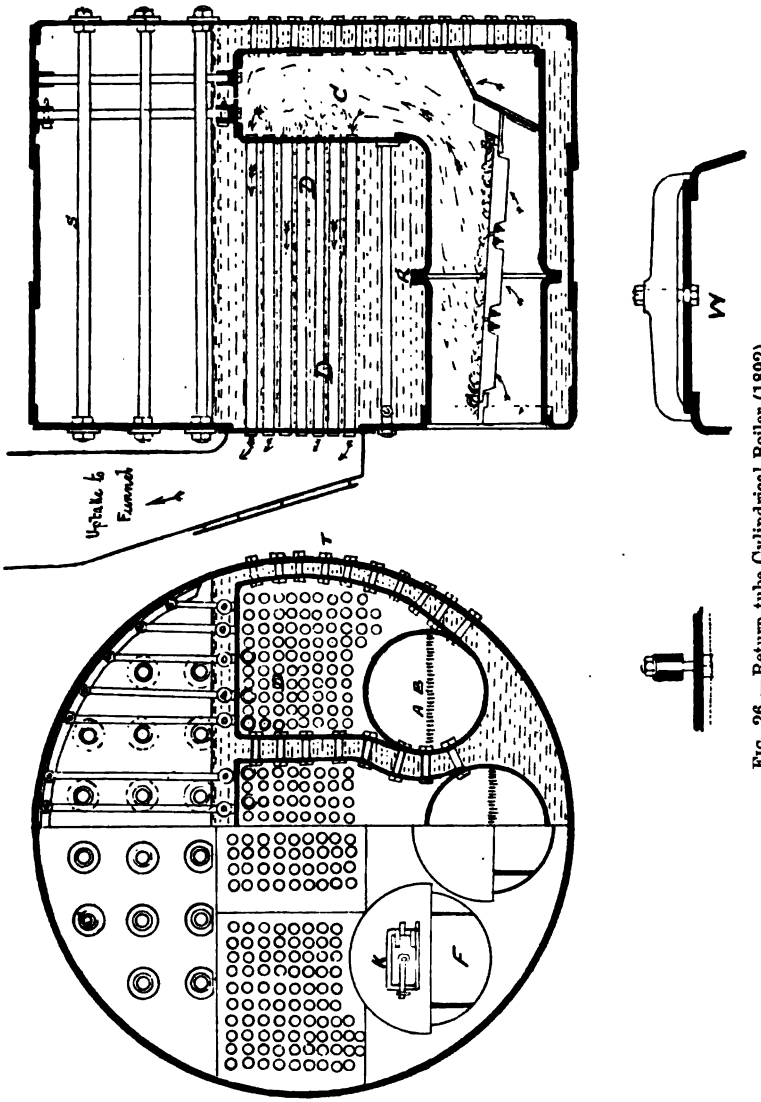


FIG. 26.—Return-tube Cylindrical Boiler (1892).

and access can be obtained to the interior of the boiler if they are arranged at convenient distances. The circular shell has been used for pressures up to 300 lb. per square inch, but, as will be noticed

later, the thickness of the plates then becomes somewhat too great for convenient manipulation in manufacture. The weight increases considerably with the pressure, rendering their adoption for general use a matter of some doubt for pressures above 220 lb., or in special cases where weight is of great importance.

In Fig. 26 a cylindrical boiler, used for moderate pressures, is shown with a front view in part section and a side view wholly in section.

There are three furnaces (in some cases four are fitted, and sometimes only one or two): *K* is a fire door, *F* is the ashpit, *AB* is a firegrate on which the fire is lighted, *C* is the combustion chamber, and *DD* the tubes leading to the uptake to the funnel. The boiler is filled with water to a few inches, generally 7 to 10, above the top of the combustion chamber, *i.e.* the highest part of the heating surface, and the space above the water-level is called the steam-room, or *steam space*. Fires are lighted on the firegrates inside the furnaces, and after they are sufficiently burnt through the fire doors are closed. The air necessary for combustion passes through the space *F* below the firegrate; a swing door, opening inwards at the bottom, is fitted at *F* to regulate the draught. The burning gases pass from the fire into the combustion chamber, and then through the tubes, internally, to the smoke box, and thence to the uptake and the funnel.

The furnaces, combustion chamber, tube plates, and tubes, which are called the *heating surfaces*, are heated by the flames and hot gases, and these in turn communicate their heat to the water, which is in contact with them all; thus steam, generated from the water, eventually fills the space above the water-level, and is compressed in the steam space until the necessary pressure is reached, when it is conducted away as required.

Circulation and Water Spaces.—Hot water is lighter than cold, and naturally tends to rise; in a cylindrical boiler, therefore, the circulation is upwards above the furnaces and other heating surfaces of the combustion chamber and of the tubes. The movement of the mixed water and steam induces a downward current at the colder parts, such as the circular shell and the front of the boiler. In the modern type of cylindrical boiler, shown in Fig. 27, it will be noticed that—

(*a*) The water spaces widen at the top, where it is possible to fit them so.

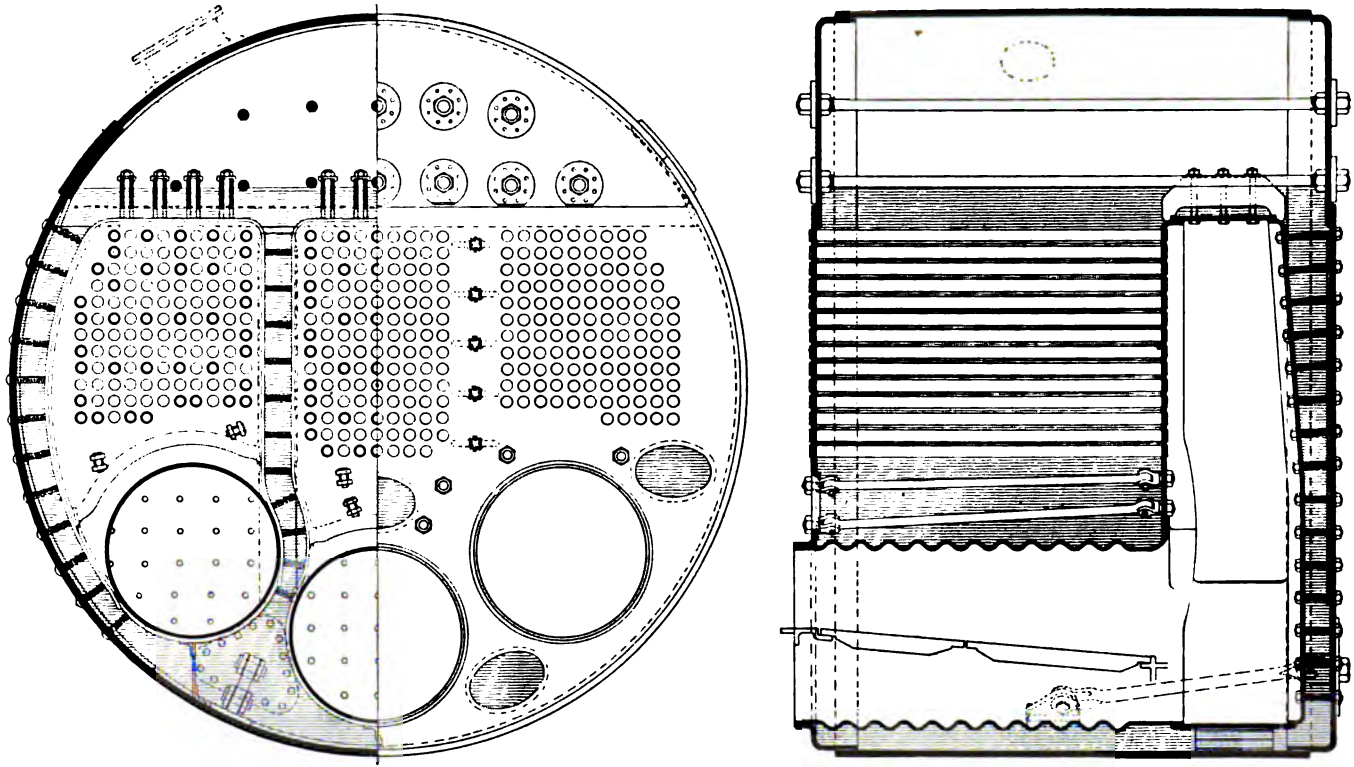


FIG. 27.—Return-tube Cylindrical Boiler (present day).

(*b*) There is a large space above the tops of the furnaces clear of tubes.

(*c*) There is a good space between the nests of tubes over each furnace.

(*d*) The tubes themselves are spaced a fair distance apart, generally 1 inch vertically and from 1 to 1¼ inches horizontally.

(*e*) The furnaces are also arranged to be as great a distance apart as is consistent with their fitting in the smallest possible shell.

Many of the early boilers used under forced draught failed because the water spaces were omitted between some of the combustion-chambers, thus restricting the circulation to such an extent that the steam could not readily escape upwards, and so caused local overheating of the tube ends and plates. In all later types there is a water-space around each combustion-chamber, and each furnace has a separate combustion-chamber. Attempts were made to separate the common combustion-chamber by brickwork; but as this does not touch the real evil, such brickwork was an entire failure and tended rather to increase than decrease the leakage. (See also Ferrules, Double-ended Boiler.)

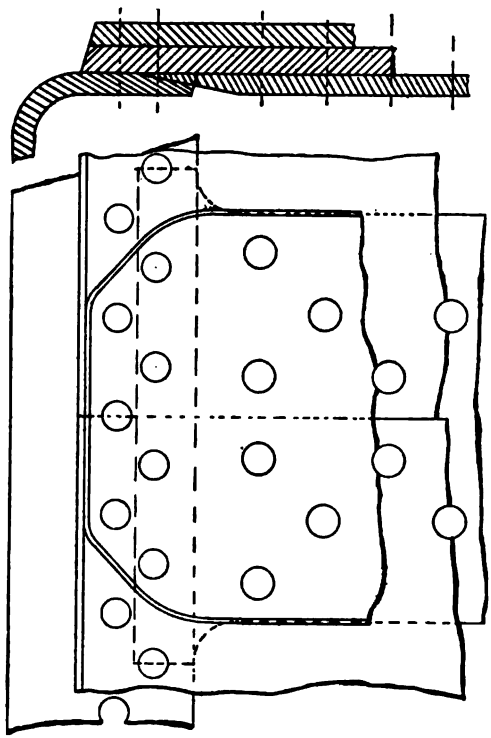


FIG. 28.—Boiler Shell Riveting.

Construction of Shell.—As seen in Fig. 27, the modern boiler is arranged with the circular plate in *one* width, with the longitudinal joints clear of the screwed stays and the boiler seating. The longitudinal seams are treble riveted, as shown in Fig. 28, with double butt straps. Double riveting is used for the junctions of the shell and end plates, and

for the junction of the separate plates of the ends. The joints of furnaces and combustion chambers are generally single riveted; this forms a good tight joint, which is nearly all that is necessary, as there is little tendency to shear at these points.

The strength at a riveted joint is less than that of the plate, because the holes in the latter, through which the rivets are passed, reduce the sectional area of the plate where the greatest stress is taken, which is along the line of rivets farthest removed from the edge of the joint, in proportion to the amount so removed. The strength of any structure is that of its weakest part, and consequently the strength of the boiler shell is governed by that of the joint employed. For the best system of riveting, the strength or efficiency of the joint is about 75 per cent of that of the plates so joined together if there are an equal number of rivets in each row, or 86 per cent if alternate rivets in the outer row are omitted, as shown in the Figure.

Shell Plates.—The shell of the boiler is subject to internal pressure, and if the circular shell be divided into two imaginary parts at any diameter, the pressure tending to cause rupture at the extremities is that of a total pressure exerted at right angles to this diameter.

Thus if P = the working pressure or load, in pounds per square inch on the safety valves;
D = the internal diameter of the shell, in inches;
L = the internal length of the shell, in inches;
t = the thickness of the shell, in inches; and
T = the tensile stress allowed on the plate, per square inch of sectional area in pounds;

then the total pressure tending to rupture the circular shell at the two extremities of any diameter—

$$= P \times D \times L;$$

and this is equal to the resistance opposed to it, which—

$$= 2 \times t \times L \times T,$$

or

$$P \times D \times L = 2 \times t \times L \times T;$$

from which—

$$t = \frac{P \times D}{2 \times T}.$$

The strength of the boiler shell to resist rupture longitudinally is greater than circumferentially. Adopting the previous notation, then the total pressure on one end of the boiler—

$$= P \times \frac{\pi \cdot D^2}{4};$$

and this pressure is distributed over the circular length of the plate, so that the resistance to rupture is—

$$= t \times \pi \cdot D \times T,$$

or $$P \times \frac{\pi \cdot D^2}{4} = t \times \pi \cdot D \times T;$$

from which $$t = \frac{P \times D}{4 \times T},$$

which is only half the thickness required to take the circumferential stress; the joints connecting the circular shell with the end plates are therefore generally quite strong enough when only doubly riveted.

Under Admiralty rules, the thickness of the shell plating of a boiler about 13 feet diameter and for a working pressure of 210 lb. per square inch, is $1\frac{1}{4}$ inches. By applying the above formulæ, it will be found that any increase in working pressure or diameter produces a large increase in thickness of plate, and consequently of weight. Until recently the manufacture of plating suitable for boiler shells was limited to a thickness of about $1\frac{1}{4}$ inches, but they are now made up to a thickness of about $1\frac{3}{4}$ inches.

Boilers constructed under the rules of the Board of Trade, Lloyd's, British Corporation, and other authorities, are slightly thicker than under Admiralty rules for the same working pressure; the principle on which the thickness of plating is calculated is, however, similar. Boilers have been constructed for working pressures up to 300 lb. per square inch. As an example of high pressures, the *Inchmarlo* has cylindrical boilers of 13 feet diameter, constructed with shell plates $1\frac{1}{3}\frac{9}{2}$ inches thick for a working pressure of 267 lb. per square inch; the thickness would be about $3\frac{1}{2}$-inch less under Admiralty rules.

Usually the thickness of the furnace plating has determined the size and working pressure rather than the thickness of the shell

plates, which can generally be adapted to any pressure which can be used for the furnace.

Furnaces.—The furnace is subjected to *external* pressure, and for the purpose of withstanding this pressure it is made circular. If the furnace is absolutely perfect in form no distortion from a uniformly distributed pressure can occur. However, such perfection is almost impossible to attain, and, consequently, in designing and manufacturing furnaces a large margin has to be allowed based on practical experience. Consider a furnace, not quite cylindrical, with its horizontal diameter somewhat greater than its vertical diameter; then the resolved vertical forces tend to produce collapse, and as the vertical diameter decreases the resolved horizontal forces also decrease, and thus reduce the resistance to collapse. This is directly opposite to the action when the boiler shell is subjected to pressure *internally*.

With low pressures the plain cylindrical furnace is strong enough to resist collapse when made of moderate thickness. An increase in thickness causes a loss in the rate of conduction of heat through the plate, but only in extreme cases is it likely to lead to overheating of the plating itself. In an ordinary boiler, from 50 to 60 per cent of the total heat absorbed by the boiler water is conducted through the furnace plates.

Various methods were employed to strengthen the furnace against collapse, and these led to the introduction of *corrugated furnaces*. The **Fox furnace** is made with a number of uniform corrugations, as shown in Fig. 27, each of which is of the same height and length; the thickness is also uniform throughout its entire length. The pitch of the corrugations is 6 inches, and the depth $2\frac{1}{2}$ inches, or a difference of 5 inches between the greatest internal and external diameter. The practical objections to the Fox furnace are that the scale accumulates at the bottom of the corrugations on the water side at the top of the furnace; the opposite side of the plate at these points is exposed to the greatest heat of the fire; and there is, in consequence of the difficulty in removing the scale, a great tendency to overheating with a reduced efficiency; the furnace also tends to elongate too much under pressure and expansion, producing unnecessary stress on the boiler plating in connection with the furnace. The increased surface area is, however, in favour of all corrugated furnaces.

The **Purves furnace**, Fig. 29, reduces the objections raised above

to a minimum, but the lack of uniformity in thickness is in itself objectionable. The pitch is 9 inches, and depth 2 inches, and the thickness at the corrugations is much greater than the remainder of the furnace plating.

In the **Morison furnace** the corrugations are pitched at 8 to 9 inches, and the depth is 2 to $2\frac{1}{4}$ inches; the plate is of uniform thickness throughout. The tendency to elongate under expansion and pressure is not so great as in the Fox type. Fig. 30 shows

SECTION OF PURVES' PATENT FURNACE IN TESTING APPARATUS.

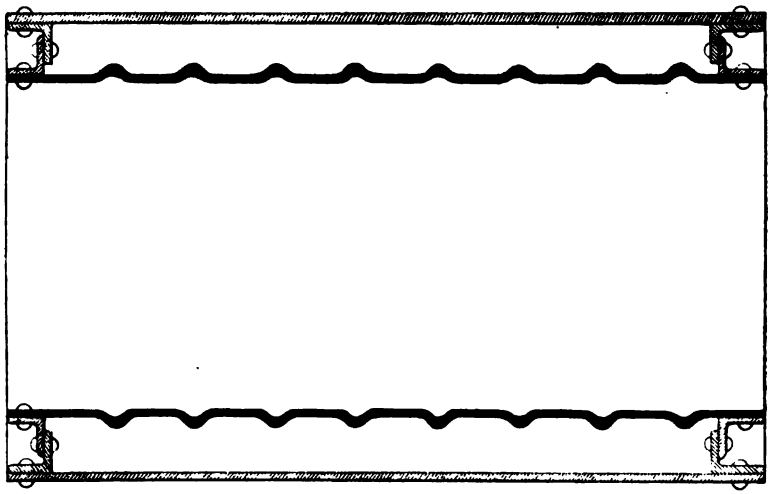


FIG. 29.—Purves Flue.

this furnace arranged so that it can be removed without disturbing any other part of the boiler structure; but this construction has for very high pressures an objection in itself—it entails a very large opening in the front plate so that the corrugations can be passed through it. The flanging at the furnace front, generally outward in modern boilers, must be of such curvature that it can be made without appreciable loss of strength. Removable furnaces entail a boiler of somewhat larger diameter for the same width of firegrate, and there is consequently an augmented thickness of shell and weight. The boiler shown in Fig. 27 is not arranged on the plan of removable furnaces, as the question of removal is not of great

importance with the greatly improved methods of manufacture and material.

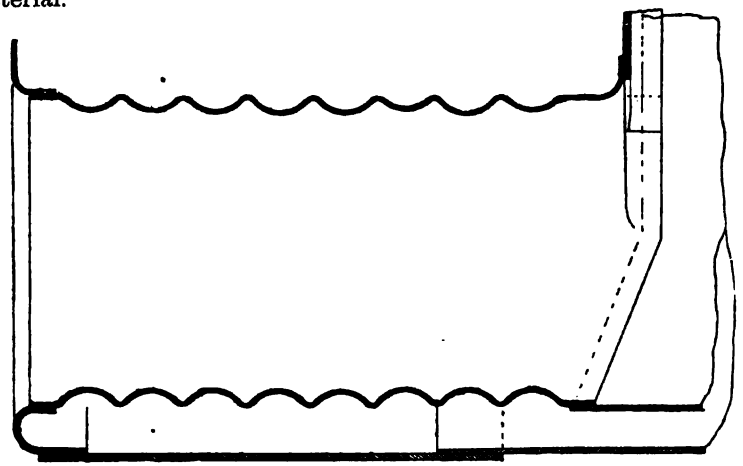


FIG. 30.—Morison Suspension Furnace.

Deighton-Morison Furnace.—This is a combination of alternate small and large corrugations, as shown in Fig. 31. The pitch of the latter is generally 8 inches, and the difference between the greatest

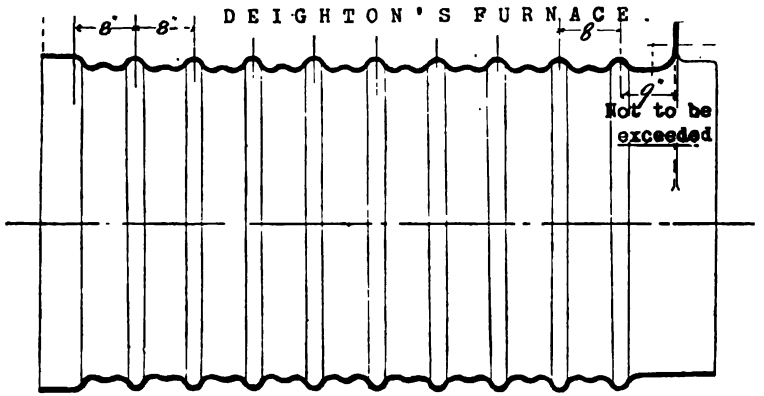


FIG. 31.—Deighton-Morison Furnace.

and least diameters is $4\frac{1}{4}$ inches. The furnace is lap welded and of uniform thickness throughout. The longitudinal elasticity is about one-third that of the Fox type, but it is sufficient for the necessary following of the expansion of the tubes. There is

about 18 per cent more heating surface than in a plain cylindrical flue.

Collapsing tests were made of Deighton furnaces in 1899 by Lloyds, with the following results:—

(1) Furnace, diameter about 32 inches, thickness $\frac{9}{16}$-inch, adapted for a working pressure of about 245 lb. per square inch; collapsed under a pressure of 1475 lb. per square inch, and the pressure stood at 1000 lb. after the collapse.

(2) Furnace, diameter about 31·9 inches, thickness $\frac{7}{16}$-inch, adapted for a working pressure of about 178 lb. per square inch;

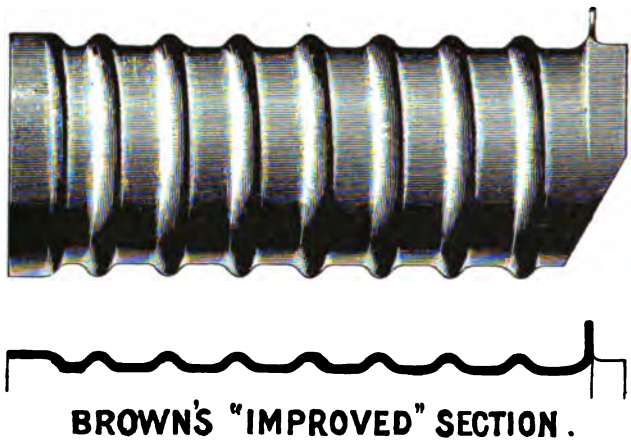


FIG. 32.—Brown's Cambered Furnace.

collapsed under a pressure of 1200 lb. per square inch, leaving the pressure standing at 800 lb.

In all cases it is usual to fit the furnaces in such a way that the corrugations of any one furnace are alternate with those adjacent to it.

Brown's Cambered Furnace.—As fitted in some warships of the *Hindustan* and *Antrim* classes, the corrugations are pitched at about 9 inches apart, and the depth is about $1\frac{5}{8}$; the plating is about 45 per cent thicker at the ridges (as in the Purves type), and the curve from ridge to ridge is similar in form to that of the Morison type, but in the "improved" type (*Lusitania*) shown in Fig. 32, the thickness is apparently uniform.

For nearly all cylindrical boiler plating, it is the common practice to fit plates about $\frac{1}{16}$-inch thicker than the specified

minimum thickness; thus in H.M.S. *Antrim's* boilers the thicknesses are: shell $1\frac{5}{16}$ inches (specified $1\frac{1}{4}$); furnace $\frac{21}{32}$ (specified $\frac{19}{32}$) at midway between ridges, and $\frac{20}{32}$-inch at ridges for which no thickness is specified. The extra thickness allows a fair margin for any inequality in manufacture and for wear. For the same reasons stays are usually larger in diameter than that specified.

Stays.—All flat surfaces subjected to fluid pressure require staying to prevent distortion. The method used for a modern boiler under high pressure is shown in Fig. 27, which is in accord with the latest developments, and the reasons for departure from the older types are detailed.

The long or *longitudinal stays* are fitted from front to back above the combustion-chambers (*S*, in Fig. 26), and an enlarged view is

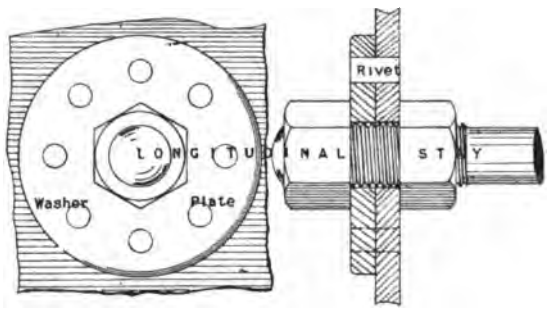


FIG. 33.—Longitudinal Stays.

given of one end of a stay in Fig. 33. These stays are about $2\frac{1}{4}$ to $2\frac{1}{2}$ inches in diameter, and pitched at a distance of about 14 to 16 inches from each other at the corners of a square. They are solid from end to end, and not allowed to be welded; at each end a *plus* thread is formed, and nuts are fitted (as shown) on both sides of the plate, with a thick riveted washer on the outer side to distribute the tension exerted by the stay over a sufficient surface to prevent local distortion. These washers are made of a diameter about two-thirds the pitch of the stays, and they are secured by rivets to the end plates; the thread on the screw is standard Whitworth, about 4 to the inch.

Above the furnaces and below the tubes there is a large unsupported space, so that staying is required. These stays are shown in detail for the end fitting in Fig. 34. They are generally made *portable*, for convenience of fitting and for removal when

cleaning the furnace and other heating surfaces. The joint of the stay is therefore made with a pin and split pin, as shown; and to allow for a certain estimated expansion, an elongated hole is made at the joint near the combustion-chamber end of the stay. The distance allowed for expansion is about $\frac{3}{32}$-inch for a length of furnace and stay of about $6\frac{1}{2}$ feet. The front tube plate between the nests of tubes has a certain unsupported area, because at the combustion-chamber end a water-space is formed between the several combustion-chambers. This space on the front plate is supported by *internal dog-stays*, as shown in Fig. 27.

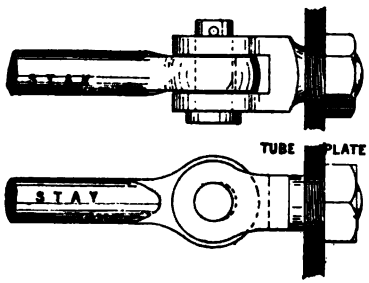


FIG. 34.—Portable Stays.

The region of the front manhole doors, near the furnaces, is

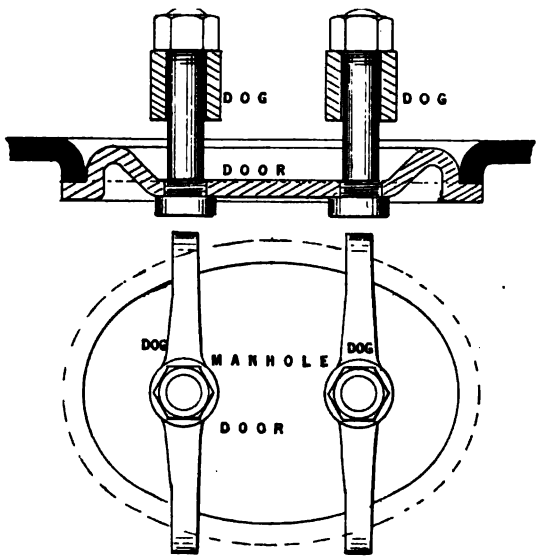


FIG. 35.—Manhole Door and Dog-stay.

stiffened by flanging the front plate internally at these places, and the flanges form elliptical rings, on which the oval doors are jointed internally, as shown in Fig. 35. The parts of the back shell plate

near the bottoms of the combustion-chambers and opposite to these manholes are supported, as shown in Fig. 27, by portable stays, jointed to a T-piece riveted to the shell. No allowance is necessary for expansion at these points, and the connections are rigid.

Combustion-Chamber and Stays.—The tops of the combustion-chambers are flat, and need staying. In Fig. 26 *vertical stays* are shown, which secure the top of the combustion-chamber to the outer shell; a forked pin-joint is fitted to each end of each stay, which is connected with an angle steel riveted to the shell, and with a screwed and nutted stay in the top plate of the combustion-chamber. An elongated hole is generally fitted to the top pin-joint to allow for difference of expansion. When steam is up, the natural expansion of the shell tends to raise it at the top, but the great pressure on

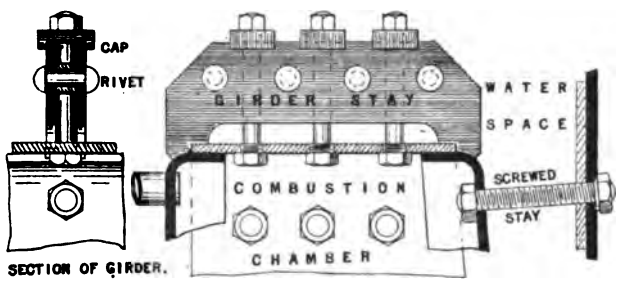


FIG. 36.—Bridge and Screwed Stays.

the top of the combustion-chamber tends to pull the shell down and to distort it. The vertical stays are not supposed to come into play until the natural expansion has taken place, but the exact allowance at the pin-joint connection is always somewhat problematical. In all recent boilers of this type, *bridge or girder stays*, which do not interfere with the natural expansion, are fitted, as shown in Figs. 27 and 36.

The *bridge or dog stays* fitted on top of the combustion-chamber are sometimes made by riveting two plates together with a distance piece or ferrule between them. The bridge is supported by the front and back plates of the combustion-chamber as much as possible. Caps are fitted on top of each bridge (Fig. 36), and stays are screwed through the top combustion-chamber plate, and nutted above the caps. The stays are made with a *plus* thread under the heads, which make steam-tight joints with the plate, and the ends of the stays are riveted over above the nuts, to prevent the latter

from slacking back; this security is also used for other nutted and screwed stays. The size and pitch of stays are in all cases calculated for the stress they have to bear.

The sides and back of each combustion-chamber form large flat surfaces, and are stayed by means of *screwed stays* nutted on each end. The stays are screwed through both the plates they secure by a continuous thread (Fig. 36), and, as far as possible, it is arranged that the stay is perpendicular to the plate exposed to the heat of the flue gases; at the other end, the plate is either sufficiently recessed or a washer fitted to form a fair surface for the face of the nut. All nuts exposed to the action of the fire, steam, or hot water are generally made of steel; but all external nuts, such as those on the ends of the long stays and on bolts of manhole doors, are made of iron.

The parts of the back shell plate opposite to the water spaces, formed by the separation of the combustion-chambers, are strengthened by means of doubling plates, and stronger stays are fitted near these parts, connecting the back plate with the combustion-chambers.

The bottoms of the combustion-chambers are made semi-circular in shape, and of rather thicker plate, so that stays are unnecessary; but an angle-steel frame, connected with the boiler shell only, supports the weight of each combustion-chamber (Fig. 27).

Tubes.—For forced-draught working the external diameter of the tubes is generally $2\frac{1}{2}$ inches, but if intended for natural-draught only they are sometimes larger. The thickness of the plain tubes is about 0·16 inch, and the ends at the smoke box are swelled out to $\frac{1}{16}$-inch greater diameter for facility of fitting and removal (Fig. 37). The stay tubes are of $\frac{1}{4}$-inch thickness, and screwed into the tube plates with a continuous thread; the front end is swelled out as with the plain tubes, and the thread is a *plus* one, so that the thickness at the bottom of the thread shall not be less than that of the tube in its plain part. The failure of some of the earlier boilers was due to the part of the tube on which the thread

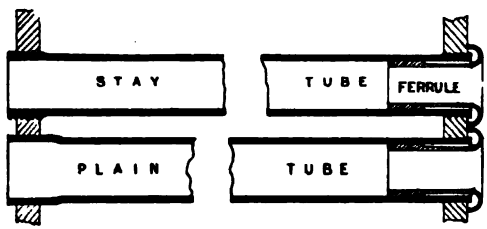


FIG. 37.—Plain and Stay Tubes—ferruled.

was cut being a separate piece welded on to the tube ends; the value as a stay was therefore very small, especially in some cases where the weld was very imperfect. The thread used for stay tubes is a fine pitched one, about 10 threads to one inch. After the tube is screwed into its place it is expanded by a tube roller in the usual way, and caulking (which would disturb its proper fitting) is not allowed in good practice. The tube ends of both plain and stay tubes should not project more than a small necessary amount, say $\frac{1}{8}$-inch, beyond the tube plate at the combustion-chamber end; this allows the cap ferrule to be fitted, as shown in Fig. 37.

In mercantile practice, *Serve tubes*, as shown in Fig. 38, are sometimes used for both plain and stay tubes. The object of their use is to increase the strength of the tube and at the same time to increase the heating surface. There are a number of ribs which extend for nearly the whole length of the tube, but which do not interfere with the usual rolling and expanding of the ends into the holes in the tube plates. The advantage gained is that for short periods a greater output of steam can be obtained from a smaller number of Serve tubes than from a larger number of ordinary tubes of the same length and diameter, but this advantage is entirely lost after the boiler has been under steam for a moderate number of hours and the tubes become choked with soot and small scoriæ. They cannot be easily swept or cleared when under steam, and for naval purposes have seldom been adopted. A more efficient method of increasing the output of the boiler is now commonly adopted in British practice, and is explained under the heading of Retarders.

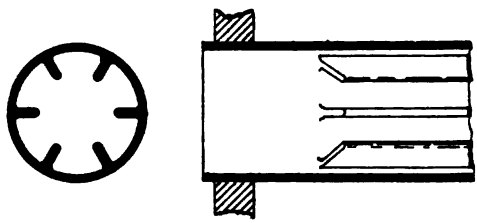


FIG. 38.—Serve Tube.

In a few water-tube boilers of the Belleville type Serve tubes are used next the fire, but only in foreign navies.

Ferrules.—The earlier boilers which gave trouble at the tube ends from leakage, under forced-draught particularly, had a ferrule fitted in the combustion-chamber end of each tube; the ferrule was simply a hollow cylindrical wedge, the outer end of which was about level with the end of the tube when driven home; it was subsequently rolled and expanded into the tube at the tube plate.

These ferrules served a very useful purpose, and in many cases kept the tubes tight when they might have failed without their assistance.

A large portion of the heat originally transferred from the tube to the tube plate was still transferred through the medium of the ferrule, and still resulted in the accumulation of steam at the surface of the plate in such quantity that overheating took place, culminating in leakage at the tube end; it mitigated, but it did not entirely cure the evil of tube leakage.

The Admiralty *cap ferrule* was then introduced, and has practically stopped all tube-end leakage within fairly high limits. The ferrule shown in Fig. 37 is made to cover the end of the tube, and to form a nearly non-conducting air-tight space with the tube plate. The heat imparted to the ferrule is conveyed along it to the part where it is in contact with the tube, about $\frac{3}{4}$-inch distant from the inner face of the tube plate; this decreases the ebullition near the tube plate, and the always limited circulation of steam and water more nearly suffices to prevent overheating. The ferrule is simply tapped into the tube end with the wooden mallet, and should not tend to distort the tube in any way.

Retarders.—The retarder consists of a piece of flat plate of the same width as the internal diameter of the tube, twisted into the form of a long spiral helix, about $2\frac{1}{4}$ turns in a length of 6 feet 6 inches; it is pushed, not forced, into the tube until its inner end is near the end of the cap ferrule, and its outer end then projects just clear of the smoke-box end of the tube. The heated gases from the fire have now to travel around this helix formation, thus bringing the whole of the tube surface into play as heating surface.

The gases take from 2 to $2\frac{1}{2}$ times as long to pass through the tubes, and thus they are given more time to diffuse their heat. The efficiency of the boiler is increased by 2 to 8 per cent by the proper use of retarders. When sweeping tubes, the retarders are removed, and this generally assists materially in keeping the inner surfaces of the tubes clean and free from accumulation of soot and small scoriæ.

Double-ended Return-tube Boiler.—When two boilers (Figs. 26 and 27) are placed back to back, with the back plates removed, and so form one structure, it is called a double-ended boiler, a type of which is shown in Fig. 39. This boiler is fired from each end, and the number of furnaces is sometimes as many as eight; but

generally six are fitted, three at each end.[1] In all recent boilers a separate combustion-chamber is fitted to each furnace, for the reasons already given; if this is done, there is no objection to the double-ended boiler as compared to the single-ended variety, and there is some saving of weight and space by its adoption. There appears, however, to be some bias in favour of the single-ended return-tube boiler, which is probably due to its more extended use with more reliable results; double-ended boilers have recently been fitted in several large passenger vessels.

H.M.S. *Medea* and *Medusa* were fitted in 1889 with boilers of

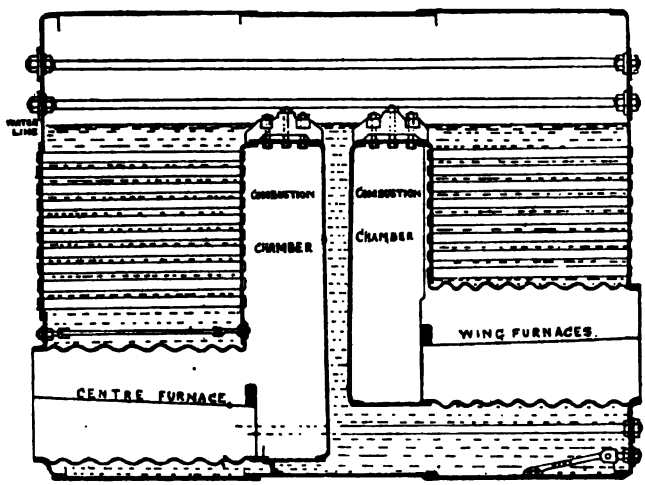


FIG. 39.—Double-ended Boiler, with Single Combustion Chamber.

this type, but with a *common combustion-chamber* to the six furnaces. This resulted in a distinct failure to realise the expected horse-power under good service conditions. Although the power was obtained for short periods on trial trips, the subsequent leakage was prohibitive of attaining it again with safety. These boilers are now replaced by the Yarrow type, with slightly curved tubes, in the *Medea*, and by the Durr type (German cruiser *Prinz Heinrich*) in the *Medusa* (since transferred to the *Vulcan*). In both cases the speed appears to be better maintained than with the original boilers.

Direct-tube Boiler.—Where the height was insufficient to fit

[1] The 25-knot Cunarders, *Lusitania* and *Mauretania*, are fitted with 23 double-ended boilers and 2 single-ended boilers, of the 4-furnace type, making 192 furnaces in all.

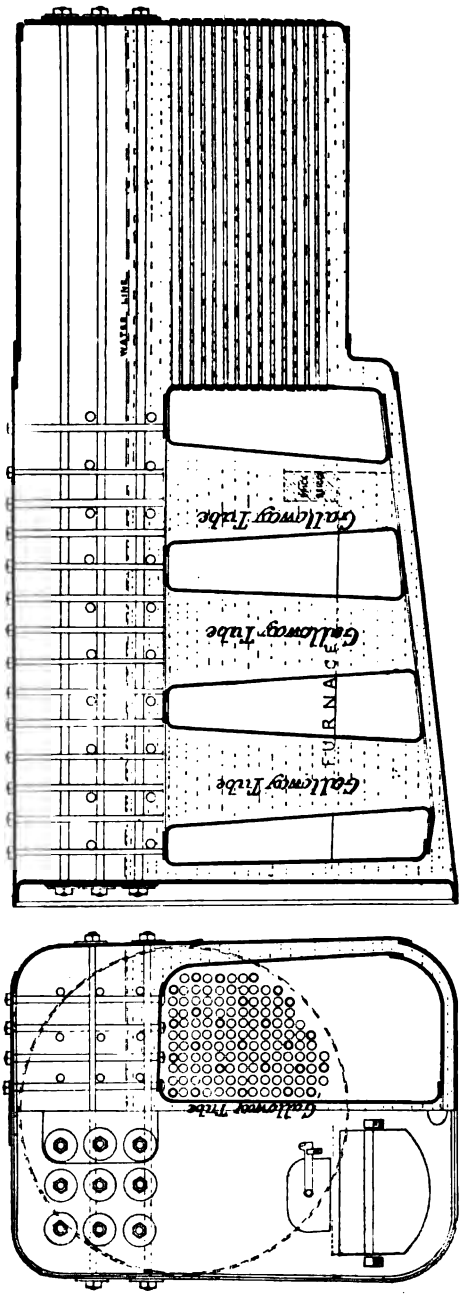


FIG. 40.—Loco-marine Boiler (*Rattlesnake*).

the return-tube boiler, the tubes were arranged in line with the furnaces instead of immediately above them; this kept the outer diameter of the boiler much smaller, and saved the height at the expense of the ground space. They were fitted in many of the earlier ships of the small cruiser type and in gunboats, and generally they were successful for this purpose; thus H.M.S. *Porpoise*, fitted with this type of boiler in 1885, was continuously employed for twenty years. The ground space and extra weight do not compete with modern types, and in all the smaller vessels for many years past upright water-tube boilers have been fitted, generally successfully.

Locomotive Boiler.—This type of boiler has been removed and replaced by water-tube boilers, except in one or two instances. It is a direct-tube boiler, and its success in locomotive practice entitles it to great respect; but the conditions of marine working are so different,

as pointed out later, that although succeeding on shore it has been a distinct failure on board ships for high rates of forcing. A boiler of this type is shown in Fig. 40; the furnace is partially divided longitudinally by Galloway tubes.

Efficiency of the Boiler.—The complete combustion of a given quantity of fuel will generate a certain quantity of heat. In practice the combustion is not quite complete, and a certain portion of the fuel passes away unconsumed, leaving only the heat generated by the remainder for possible transmission to the water in the boiler through the heating surfaces.

The *efficiency of combustion* is, therefore, the ratio of the heat available for transmission to that contained in the fuel.

Of the heat available for transmission, only a certain amount is actually transferred to the water in the boiler; the remainder passes away, and is lost principally up the funnel.

The *efficiency of the heating surface* is the ratio of the actual heat transmitted to the water to that available for transmission.

The *efficiency of the boiler* is the product of these two efficiencies of combustion and of the heating surface; or it may be given as the ratio of the heat actually transmitted to the water to that contained in the fuel.

Efficiency of the boiler

= efficiency of combustion × efficiency of heating surface.

$$= \frac{\text{heat available for transmission}}{\text{heat energy of fuel}} \times \frac{\text{heat transmitted}}{\text{heat available for transmission}}$$

$$= \frac{\text{heat transmitted to water}}{\text{heat energy of fuel}}.$$

Details of some recently constructed Cylindrical Boilers

	Date of Design.							
	1895.	1892.	1895.	1900. (ss. *Saxonia*).	1902. H.M.S. *King Edward VII.*	1903. H.M.S. *Africa.*	1904. H.M.S. *Triumph.*	1907. *Shannon.*
Pressure — boilers	155	155	155	210	210	210	275	275
Pressure — engines	150	150	150	205	205	205	250	250
Shell — diameter	16′3″	16′0″	14′0″ (?)	14′0″ (?)	13′0″	15′5″	Yarrow	Yarrow
Shell — length	9′6″	18′0″	9′11″	9′0″	9′1″	10′7″		
Shell — thickness	$1\frac{7}{32}$″	$1\frac{3}{16}$″	$1\frac{1}{32}$″	$1\frac{1}{2}$″	$1\frac{1}{4}$″	$1\frac{7}{16}$″		
Furnaces — number	4	8	3	3	3	3	12 boilers	24 boilers
Furnaces — diameter	45″	45″	46¼″	46″	38″	49⅜″		
Furnaces — length	7′0″	7′0″	7′0″	5′6″	5′9½″	6′9″		
Furnaces — thickness	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{1}{2}$″	$\frac{11}{16}$″	$\frac{13}{16}$″	$\frac{3}{4}$″		
Grate surface (sq. ft.)	817	812	649	570	324 (for 6 boilers)	243 (for 3 boilers)	664	1358
Heating surface (sq. ft.)	25,233	24,908	18,464	25,803	10,158	8328	37,524	80,424
I.H.P. — F.D.	13,000	12,000	9600	9000 Howden's system	5000	3600	12,500	27,000
I.H.P. — N.D.	9000	10,000	8000	…	4000	2880	…	…
Coal per I.H.P. (lb.) — F.D.	2·40	2·3	2·42	1·29	1·67	1·76	1·73	2·0[1]
Coal per I.H.P. (lb.) — N.D.	2·25	1·79	1·74	…	1·74	…	…	…
Weight in tons	724	646	557·4	910 (1100) total estimated	…	…	440	2410[2]
I.H.P. per ton — F.D.	16·6	18·6	17·5	8·2	…	…	28·3	11·2[2]
I.H.P. per ton — N.D.	13·9	15·5	14·6	…	…	…	…	…
I.H.P. per sq. ft. grate — F.D.	14·7	14·8	15·4	17·6	15·5	15·0	18·8	19·9
I.H.P. per sq. ft. grate — N.D.	12·2	12·3	12·8	…	12·3	12·0	…	…
Heating surface per I.H.P. — F.D.	2·10	2·08	1·94	2·85	2·03	2·3	3·0	2·98
Heating surface per I.H.P. — N.D.	2·52	2·49	2·32	…	2·54	2·75	…	…
Floor space per I.H.P. — F.D.	0·328	0·288	0·321	…	…	…	…	0·27
Floor space per I.H.P. — N.D.	0·393	0·345	0·385	…	…	…	…	…

Notes.—The surfaces are given in square feet. The weights include all boiler-room weights. The floor space includes all boiler-room spaces.

The Yarrow boiler is included for purposes of comparison.

[1] Estimated from the usual practice; probably 1¾ on trial.

[2] Total weight of machinery and boilers.

Weight of Cylindrical Boilers.—Until about 1890 there was a gradual decrease in weight of boiler per I.H.P., because the use of forced draught and better materials, and the increased economy obtained in the engine by using higher working pressures, counterbalanced the increase of weight necessary for the proper strength of the boiler and engine.

When the working pressure was increased from 150 or 160 lb.

to 210 lb., the weight of cylindrical boilers increased with the pressure. With these boilers the weight increases nearly in proportion to that of the shell plating, or—

$$\text{Weight} \propto t \propto \frac{P \cdot D}{2 \cdot T};$$

and for boilers of the same diameter D and strength T of material—

$$t \propto P.$$

Thus, if the pressure be increased from 200 to 300 lb. per square inch, the increase in weight of shell

$$= \frac{300 - 200}{200} = \frac{100}{200} = 50 \text{ per cent};$$

but the fittings, casings, and other parts are the same for both installations, and their weight is about 60 per cent of the total weight included in that of the boiler rooms; so that the actual increase is—

$$\frac{100 + 50 + 60 - 100 - 60}{100 + 50 + 60} = \frac{50}{210} = 24 \text{ per cent nearly.}$$

Although the size of engine is decreased with increased pressure there is a slight increase in weight, as greater strength is necessary for all parts through which the steam passes.

There are various ways in which boilers can be compared in relation to their weight:—

(1) *Weight of water evaporated* per hour per ton weight of boiler.—If the rates of combustion be equal, the comparison is a very fair one; but proper means must be taken to ascertain the quality of the steam on leaving the boiler in regard to priming, or degree of moisture carried away in the steam.

(2) *Grate surface per ton* of weight.—There may be considerable difference in the length of grate; thus for the mercantile marine it may be only 5 feet or less, whereas in a warship it is sometimes 7 feet or more. A short grate is generally more efficient.

(3) *Heating surface per ton* of weight.—The fairest comparison can be made on the heating surface provided per ton of total weight, including water at working height in the boiler. Some parts of the heating surface are more efficient than others, but for

general purposes of comparison this does not materially affect the result. With a moderate rate of combustion the efficiency of the heating surface (and relatively, the boiler) is generally greater than with a very high rate.

(4) *I.H.P. per ton* of weight.—This is a fair comparison only when the rate of combustion and the weight of steam used per I.H.P. in the engine are the same for each plant compared. The rate of combustion varies from about 15 lb. per square foot of grate in a tramp steamer to 220 lb. in an express locomotive, while the weight of steam used per I.H.P. varies from 11 lb. per hour in a few of the most efficient steamers to about 20 lb. in a destroyer. The most recent cruisers and battleships use about 17·5 lb. weight of steam for *all purposes*, of which about 15 lb. are for the main engines, per I.H.P. per hour.

CHAPTER VII

UPRIGHT WATER-TUBE BOILERS

Circulation—Above Water Delivery.—The circulation caused by convection in boilers with below water delivery has already been explained in Chapter IV. The Belleville generator, the Niclausse boiler, the early Thornycroft boilers, and to some extent

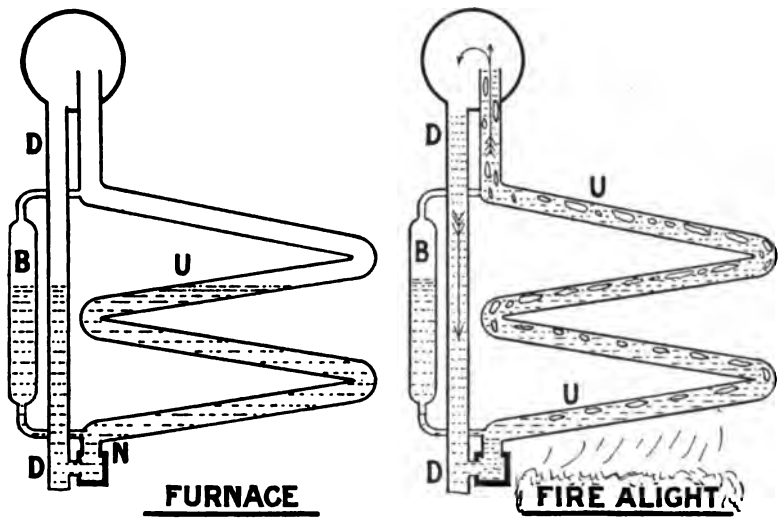


FIG. 42.—Circulation : Belleville Generator.

the Babcock and other boilers, are so arranged that the mixed steam and water coming from the generating tubes are discharged into the steam chest above the normal working water-level.

In the Figs. 42 and 43, the water-level, *when cold*, is shown in outline diagrams of the Belleville and Thornycroft (old) types. After the fire is lighted the water in the tubes U becomes gradually

hotter and expands. This causes an apparent rise of water-level in U; but as the weight (and, therefore, the head of pressure) is not increased, the water-level in D, which is little exposed to heat, remains unchanged. Steam bubbles soon begin to form in the water in the tubes U, and push the apparent water-level in them higher and higher until an overflow into the steam chest takes place. Up to this point no change has taken place in the level of water in D, because no change of head of pressure has taken place in either tube. At the instant that a discharge from U into the steam space takes place, the balance of head is upset; that in U is decreased, and that in D is increased by any water discharged into it from U. (Any steam which is formed accumulates in the upper part of the steam chest, and such pressure that it may exert is equally supported by the water in D and in U.) The greater head created in D causes a movement from D to U, and starts a circulation which is continuous in direction, although the discharge of steam and water from the tubes U into the steam chest is generally intermittent.

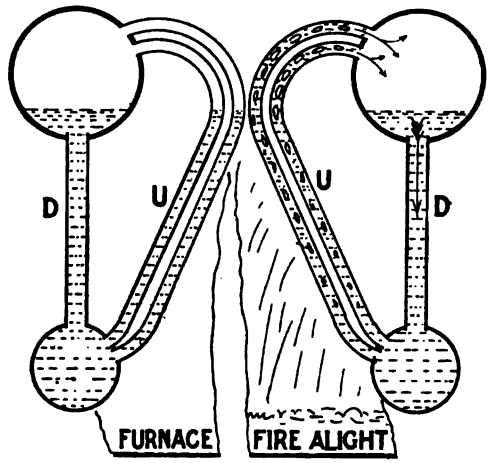


FIG. 43.—Circulation: Thornycroft (old type) Boiler.

In the Thornycroft boiler the level of the water in D remains practically constant, and is that of the working water-level. In the Belleville, however, it is probable that the level in D rises until it reaches the bottom of the steam chest; with this change in level, there is an increase of difference of head.

In the Thornycroft boiler the circulation is free and unrestricted; there is only one length of tube (from 10 to 15 feet long), and no acute changes in direction. The circulation is therefore rapid although the difference of head is small; from model experiments, Sir John Thornycroft calculated that each particle of water passed through the steam chest 105 times before leaving it as steam.

In the Belleville boiler the frictional resistance to the circula-

tion is great, owing to the nearly horizontal placing of the tubes—there are also about 70 to 80 feet of length of tube with about fourteen to sixteen acute changes in direction,—and the circulation is restricted by the small passages N at the nipple connections. The circulation is therefore comparatively slow, and to prevent probable overheating of the metal of the tubes the safe rate of combustion for sea-going purposes should not exceed 30 lb. per hour per 1 sq. ft. of grate, or about 1 lb. per 1 sq. ft. of total heating surface. A non-return valve is also fitted to each downcomer D because, when first lighting fires, there is a possibility of circulation starting in the wrong direction; this would be disastrous, as a supply of water cannot be maintained owing to the nature of the construction of the boiler. These non-return valves are also fitted for another purpose, namely, to equalise the supply when more than one downcomer is fitted.

In all cases circulation, when once started in any one direction, tends to continue in the same direction; this is due to convection and change in density (as described above) producing difference of head; to momentum caused by this difference; to further increases of head and momentum caused by discharge of water from the generating tubes into the steam chest; and to the (kinetic) energy of motion produced by the momentum (force).

Downcomers.—All boilers built for the Navy are fitted with separate downcomer tubes, of fairly large diameter and remote from the fire and intense heat. They are generally fitted outside the casing enclosing the generator tubes. For a large boiler two downcomers of about $4\frac{1}{2}$ inches diameter are usually fitted to each water drum, either back or front, as convenient. Downcomers are necessary for marine purposes, when the ship is rolling or pitching; but on shore they might be safely dispensed with for boilers of the Yarrow, White-Forster, and Normand types.

If all the generating tubes are curved the downcomers may be straight, because expansion is thus allowed independently; but if the tubes are straight, as in some Yarrow boilers, the downcomers should be curved about $\frac{1}{4}$-inch in each foot of length.

Advantages of Water-tube Boilers.—Some advantages are claimed for water-tube boilers compared with the ordinary cylindrical marine type. These are briefly considered as below:—

1. *Strength.*—If sufficient weight is allowed, it is now possible to build a boiler of any type for any pressure up to about 360 lb.

per square inch. If the weight is limited the strongest boiler is one of the water-tube variety, and with which the smallest tubes can be used with safety.

2. *Safety.* — Comparatively a very small weight of water is carried in a water-tube boiler at working height, and its sudden release, by fracture, is likely to be less disastrous than a larger quantity. The weight of water carried varies, according to type and size of tubes, from about one-tenth to one-fourth that of a cylindrical boiler of equal power.

3. *Lightness.*—The smaller quantity of water carried at working height, and the absence of any heavy shell, reduce the weight of the boiler, power for power. There is a great difference in the weight of water-tube boilers of various types, but generally the lightest boiler is that with tubes of the smallest diameter which can be safely employed.

4. *High Power.*—The power which can be obtained from a boiler depends on the area of heating surface which can be provided within a limited space and weight, and the greatest area of heating surface is obtained by using water-tube boilers with tubes of the smallest permissible diameter. A larger proportion of the ground space occupied by a water-tube boiler can be utilised as a firegrate, and, consequently, the rate of combustion need not be so high, power for power and space for space, as in a cylindrical boiler, and a higher percentage of the full power can be maintained for longer periods with a lower rate of combustion, because the fires do not require clinkering so frequently.

5. *Facility of Repair.*—Permanent defects are frequently produced by sudden alterations of temperature in cylindrical boilers. The water-tube boiler is not usually affected in this way, as some provision is made for independent expansion of the various parts. (Frequent and undue haste in cooling is not conducive to the long life of any boiler; statements made in the opposite sense must be considered as comparative, as pointed out at the beginning of this section, and not absolute.) Fires can, therefore, be withdrawn and the steam and water blown out without any delay in cooling the plant, and any defects made good. A tube or element can be made good in a comparatively short time, and, in most of the boilers with upright tubes, a tube can be plugged in only a few minutes. The defects usually met with in cylindrical boilers are of a serious nature, and require considerable time to make good.

6. *Durability.*—As the generating tubes constitute the principal part of a water-tube boiler, its life practically depends on the life of the tubes. So far as present experience goes, the life of the tubes in a water-tube boiler is greater than of those in a cylindrical boiler, when used with forced draught of sufficient intensity to produce the same power on the same weight.

The life of the tubes depends on :—

(*a*) The thickness allowed for wear, especially of the tubes in the rows next the fire.

(*b*) The inclination of the tubes and the rate of circulation through them. An upright tube, with a comparatively rapid circulation, is less liable to overheating, and consequent reduction in material and strength, than a horizontal one. The deposit of scale is lessened by a quick circulation.

(*c*) The purity of the water used for feeding the boiler, and its freedom from grease and acidity.

(*d*) The class of material, and the treatment of those in charge.

The generating tubes in destroyers and small vessels, where weight is of primary importance, are thinner than in other and larger vessels, and consequently their life is shorter. Even in these small vessels the margin for wear and tear is considerable, and instances are known where the decrease in thickness has been difficult to determine after some years' service. For examination a few tubes are withdrawn and thoroughly cleaned and scraped inside and outside; they are then compared by weight with new tubes of the same length and thickness. Further examination is made by splitting and cutting up a certain number of tubes into short lengths, and then carefully gauging them.

From the foregoing it is evident that the upright tube boiler can be made more durable than the nearly horizontal type, and that it possesses some tangible advantages, such as saving in weight, economy, and in many instances simplicity. As far as can be seen at present, a further development in upright tube boilers is very probable for both naval and mercantile purposes.

7. *Handiness.*—The furnaces in water-tube boilers are all of one convenient height for stoking. In a cylindrical boiler the heights differ by $1\frac{1}{2}$ to $2\frac{1}{2}$ feet. Steam can be raised quickly, and the time required for raising steam is only limited by that required for warming the engines. If the engines are already moving, more boilers can be quickly added to those already in use.

The rapidity with which steam can be raised depends on the quantity of water which must first be heated to produce steam, and in water-tube boilers this quantity is considerably less than in the water-tank type.

8. *Economy of Fuel.*—In time of war, ships will frequently be cruising at moderate speeds, and the higher speeds will only be required at uncertain intervals, at short notice. High speed at short notice necessitates keeping all, or nearly all the fires in cylindrical boilers alight or banked, and a consequent steady diminishing of the supply of fuel. It is unnecessary to keep the fires banked in boilers in which steam can be raised quickly, and the fuel otherwise burnt is thus saved, and the steaming distance and period which a ship can keep the sea, without fresh supplies of fuel, are thus greatly increased.

"In the war with Russia it was found that Japanese ships with cylindrical boilers consumed *five* times the coal burned by those with water-tube generators, owing to the fact that, having to be ready at two hours' notice, they had to keep fires going while the Belleville boilered ships were able to let fires out."—*Heresies of Sea Power*, by Fred. T. Jane, 1906.

Generally, and except in cases of extreme urgency, fires should *not* be banked, but allowed to burn down for from six to eight hours, and then just kept alight near the furnace door and ready for pushing back; the boiler should not be cooled suddenly or quickly. About 14 cwts. to 1 ton of coal is required for raising steam in each boiler, and when banked at one hour's notice about 2 cwts. of coal is required per hour, and about double this quantity at short notice. It is therefore more economical to let the fires burn out, if steam is not probably required for about twelve hours, and when about one hour's notice can be given. During this time the cleaning of the fires and furnaces, sweeping tubes, possibly minor repairs, and preparation for steaming by re-laying the fires, may be carried on so that the ship is ready for full power, as opposed to something less with dirty fires and heating surfaces.

With, say, one-half the boilers alight and engines moving slowly, and therefore warmed through, a ship should be able to obtain in a few minutes nearly four-fifths of her maximum speed; but to maintain this speed more boilers must be added to allow for cleaning fires, etc. It appears, then, that it is more economical, and conducive to obtaining full power, to keep only a few boilers under

steam or banked at short notice, and the remainder in all respects clean and ready, but with fires not alight. It is generally more economical, when high speeds are required at very short notice, to keep the engines moving slowly than to move them from time to time; this is because it requires some time to warm the engines through, and because the safe conduct of the operation of adding fresh boilers, especially in times of urgency and emergency, should be controlled by one head. Celerity with safety cannot be attained by drill in matters of boiler or machinery control, but both are possible by a cool and experienced head. "Move with caution but move promptly" is, perhaps, the keynote of successful practice; but promptness without caution produces more accidents and makes more defects than any other cause.

WHITE-FORSTER BOILER

FIGS. 44 and 45.

Typical Features.—There are two water-chambers and one steam collector. All the tubes are drowned—that is, they deliver their contents below the working water-level. The radius of curvature of each tube is the same, and the curvature is only sufficient to determine the direction of movement when expanded by heat. The tubes are arranged like the staves of a barrel, and the continuation of the line of curvature of each tube, passing through the steam chest, allows each tube to be withdrawn into and from the steam chest through a manhole. Each tube can therefore be withdrawn without disturbing any other tube. The tubes can be cleaned internally, as shown in the figure, by a tube brush whose rigid handle is of the proper curvature. This system of cleaning is probably the best yet devised.

Circulation.—Large downtake tubes are provided at the back end of the boiler, and the circulation is unrestricted. No separator plates are necessary, and only the usual internal steam-pipe is fitted. A feed regulator is generally fitted to each boiler.

Path of the Furnace Gases.—The uptake is divided near the tubes by patent baffles, as shown in the figure, which have the effect of drawing gases equally over the whole of the tube surface, and at right angles to it. (The grips in the Yarrow boiler have a similar effect, but this arrangement is simpler.) The number of baffles is arranged to suit the length of the tubes and

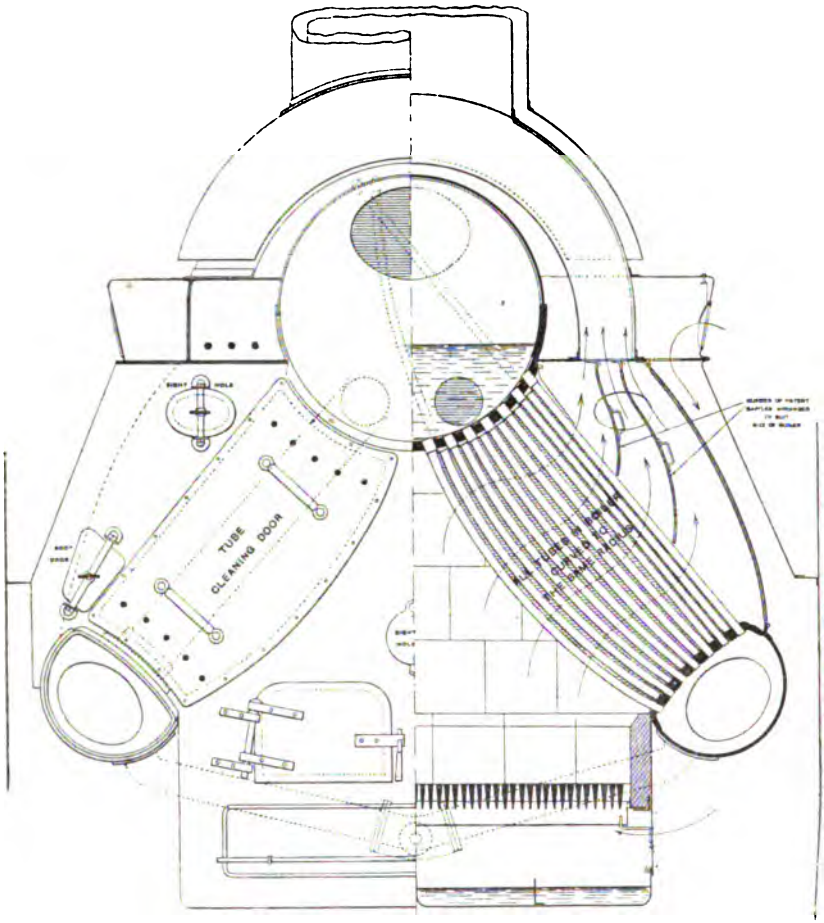


FIG. 44.—White-Forster Boiler.

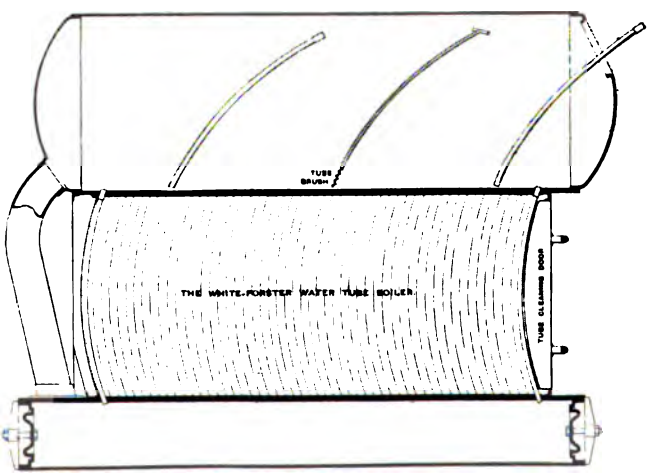


FIG. 45

conditions of working. Large doors are fitted at the stokehold end, so that the whole length of tubes is exposed for cleaning, and a brush can be passed from end to end of the boiler between the rows of tubes while the fires are alight.

Remarks. — The good features of this boiler are: the slight curvature of the tubes, the facility of withdrawal and renewal of the tubes, and the practicability of cleaning the tubes, both externally and internally. The boiler is sometimes constructed to be fired from opposite ends, and is known as the double-ended type.

YARROW BOILER

FIG. 46.

Typical Features.—The generating tubes are fitted between one upper steam chest and two lower water-chambers. The tubes are inclined at any convenient angle to suit the proposed height of boiler and width of firegrate, and generally the inclination lies between 45° and 60° to the horizontal. The tubes, when first fitted, are usually straight; but, after steaming, are generally found curved, to a more or less extent, by the action of the fire and difference of expansion. The continual change of form does not conduce to durability, and in boilers which are subjected to high rates of forcing it is now usual, for naval purposes, to make the two rows of tubes next the fire with sufficient curvature to allow for independent expansion.

Circulation. — When downcomers are fitted, the course of circulation is similar to that of other boilers with downcomers. In boilers in which no down tubes are fitted the circulation is various, because at different rates of combustion, or conditions of rolling or pitching, a tube may be either an uptake or a downtake under the altered conditions. On shore, when the boiler is perfectly steady, the circulation, having once started up the tubes nearest the fire, and down some of those more remote from the fire, continues in the same direction, and its rate of movement accommodates itself to the necessary conditions.

In a few cases, instead of fitting large outside downcomers, some of the small tubes are made to act as downcomers by forming pockets with thin baffle plates inside the water-chambers. If the tubes in the wings, remote from the fire, are thus screened off, the supply may be temporarily stopped by the ship heeling; and if a

few tubes at one end are used as downcomers, the allowance for expansion, between these comparatively cool tubes and the others acting as steam generators, is inadequate when all the tubes are straight.

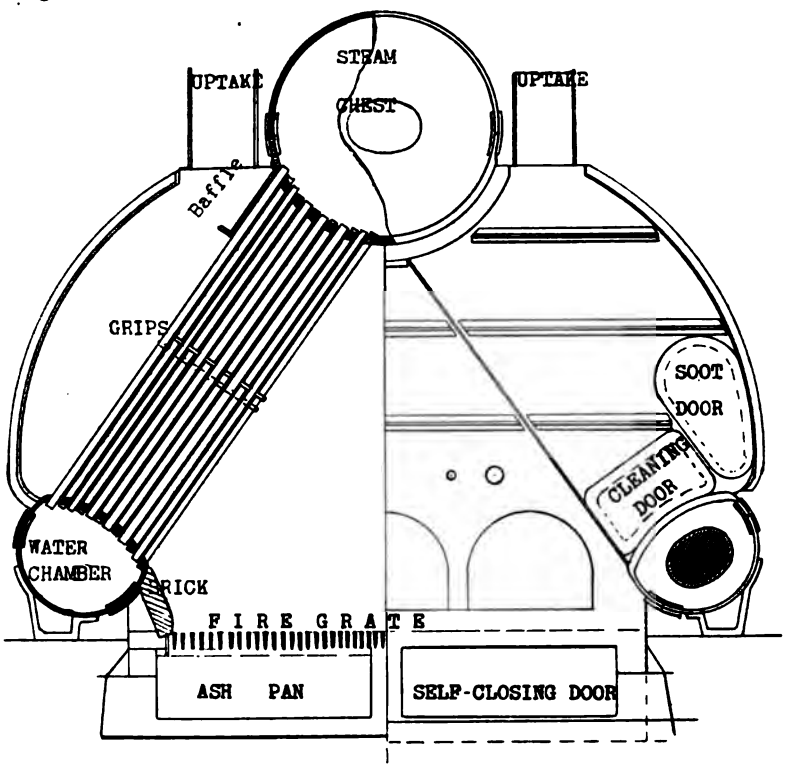


FIG. 46.—Yarrow Boiler.

Path of the Furnace Gases.—About 18 inches from the top of the tubes on the outer sides a hanging baffle is fitted to prevent a too easy run to the uptake. At about the mid-length, or at other intervals, grips are fitted diagonally between the tubes and at right angles to them. These grips form baffles, and deflect the gases across the tubes, leaving no spaces untouched by a fair proportion of the heat. The whole distance traversed by the gases among the tubes is less than 3 feet, and little time is therefore allowed to impart heat, especially if the fires are forced; but this does not appear to adversely affect the efficiency of the boiler.

Remarks.—The air supplied for combustion is sometimes heated by passing it through an air casing, outside the boiler casing and self-closing doors, to the fires and ashpits. The Yarrow boilers fitted, in conjunction with cylindrical boilers, in some of the recent cruisers, are adapted for the Howden system of closed ashpits and hot-air supply, and some considerable economy is thus obtained.

In the *Triumph* (see Fig. 105) no downcomers are fitted, and the tubes are of a uniform external diameter of $1\frac{3}{4}$ inches. Each boiler, of about 1200 I.H.P., contains about $5\frac{1}{2}$ tons of water at working height, and it would, under normal conditions of full-power working, take about thirty-six minutes to evaporate this amount. No feed regulator is fitted, and is stated to be unnecessary; but it undoubtedly adds to the convenience and ease of working, and regulators are fitted to Yarrow boilers of the *Antrim* and more recent vessels. Some tests made on shore showed that by decreasing the grate area from 53 to 40 square feet (length of grate from 6 feet 10 inches to 5 feet $9\frac{1}{2}$ inches), with an increased air-draught from 0·56 to 0·75 inch, a better economy was obtained. This is extremely probable, as it is well known that a grate of over 6 feet in length cannot be very efficiently stoked.

The holes for the reception of the tubes are first drilled through the tube plates, then a harbouring tool is used to drill away a portion of the plate in the wake of the holes, so that each tube shall be in contact only with a right-angled cylindrical surface of moderate length, as shown in Fig. 54. After the tubes are in place they are expanded by a tube roller, in the usual way, to make them tight in their holes, and then the ends are bell-mouthed with a pneumatic hammer. The withdrawal force, or total unbalanced axial tension in each tube, is about 700 lb. for a pressure of 270 lb. per square inch. Some experiments with a tube for the *Triumph* showed that it required a force of 2 tons to stretch the tube about $\frac{1}{64}$-inch, and that withdrawal only took place after stretching the tube $5\frac{5}{8}$ inches with a pull of $13\frac{1}{2}$ tons. Boilers of this type, but *double-ended*, are being built for a foreign power (1908).

Hawthorn Boiler.—The *Amethyst* and a number of destroyers, including the *Eden* with steam turbines, are fitted with boilers made by Hawthorn, Leslie, & Co. The majority of the tubes are straight, as in the Yarrow type, while the two rows next the fire are bent towards the fire, as shown in Fig. 47. The curvature is

regular and about $\frac{5}{8}$-inch to each foot of length. There are two downcomers at the back end, one to each water-chamber. The boiler mountings are of the usual type, except that the feed water is delivered from the feed pump through a pipe with two branches leading to the respective *water-chambers*, with a check valve on each. (This is a notable, and perhaps singular, exception to the general rule that the feed water is delivered into the steam chest.) An economiser is formed by separating the three rows of tubes more remote from the fire on each side from the remainder by a pocket in each (lower) water-chamber. Inside this pocket is the internal

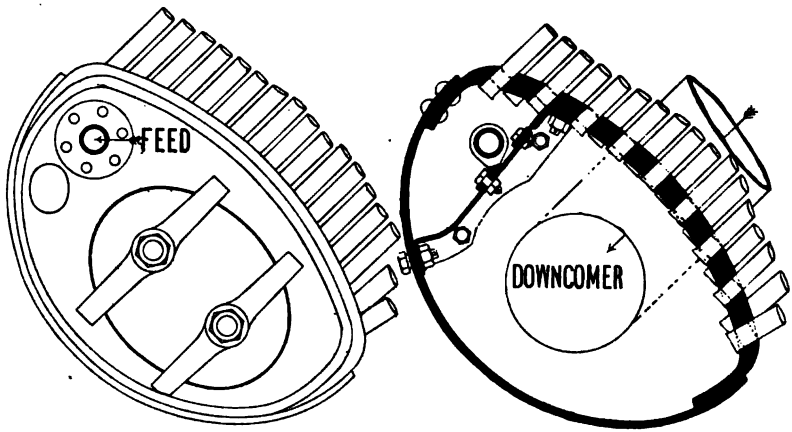


FIG. 47.—Hawthorn Boiler, with Economiser.

feed pipe, made of galvanised steel, with holes perforated in it and with its extreme end plugged.

The *circulation* is from the internal feed pipe up through the economiser tubes, downwards through the large downcomers, and upwards through the generating tubes; being delivered into the steam chest as part steam and part water. Rather over 20 per cent, or one-fifth, of the total heating surface of the boiler is used for feed heating. The economiser is in the correct position, and tends to reduce the funnel loss by decreasing the temperature of the gases. The construction of a pocket in each water-chamber is inconvenient, but compares favourably with, say, the Belleville type with regard to simplicity. A small sight-hole is made just below the feed valve for examination and ready removal of scale; but a thorough cleaning can only be made by removing the plating forming the pocket, which

is conveniently arranged for easy detachment. No baffles are fitted among the tubes, but one is fitted in the wake of the uptake, as in the Yarrow boiler. The row of tubes next the fire is incomplete, and each alternate pair of tubes, of the otherwise complete row, is placed in the third row; this third row is therefore formed of alternate $1\frac{1}{8}''$ and $1\frac{3}{8}''$ tubes in destroyer boilers. The advantage of this arrangement is not apparent.

THORNYCROFT-SCHULZ BOILER

FIGS. 48 and 49.

Typical Features.—There are three water-chambers and one

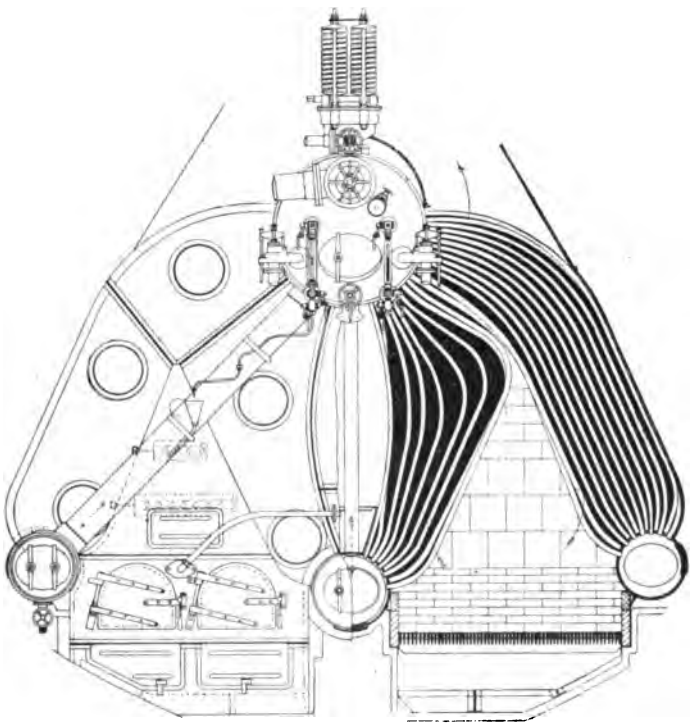


FIG. 48.—Thornycroft-Schulz Boiler.

steam chest. Nearly the whole of the tubes deliver their contents of steam and water below the normal working water-level. The curvature of the tubes is generally moderate, and in the various

types slight differences in the arrangement of the water-walls are made. The water-walls are indicated in the figures by two tubes partially overlapping each other.

Circulation.—In both Figs. 48 and 49 each wing water-chamber is supplied with water from the steam chest by a single large downcomer pipe, which is fitted outside the boiler casing at the stokehold end. The boilers for British ships are fitted with several large downcomers, immediately above the middle water-chamber, as shown in Figs 48 and 56. For German vessels a number of small tubes, similar to the generating tubes, and shielded by water-walls, sometimes take the place of the large downcomers to the middle water-chamber, as shown in Fig. 49. In some others of the German type the supply to the wing water-chambers is carried through nearly horizontal pipes from the middle chamber, as in the *Daring* type (Fig. 56). Thornycroft feed regulators are usually fitted to each boiler.

Path of the Furnace Gases.—Fig. 48 shows a front elevation, in part section, of a Thornycroft-Schulz boiler as fitted in some recent naval vessels. The gases, rising from the fires, pass among tubes rising from both the middle and wing water-chambers, as shown by the arrow heads, and escape to the uptake on both sides of the steam collector. The uptake, usually near the back end, rises above the boiler, and thus a large portion of the gases, traversing the tubes nearly horizontally, rises gradually, and not directly, into the uptake.

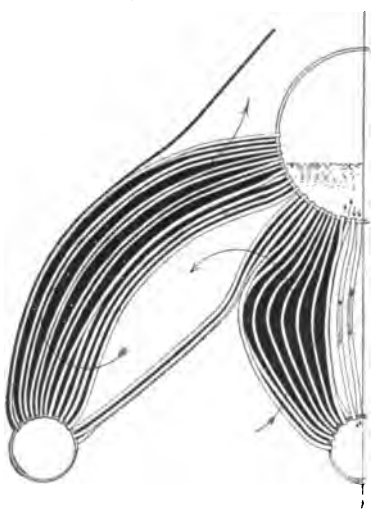


FIG. 49.—Schulz-Thornycroft Boiler.

Fig. 49 shows the arrangement of the tubes and water-walls as fitted in some vessels of the Imperial German Navy. A second, or auxiliary, combustion-chamber is formed among the wing tubes. The course of the furnace gases is shown by the arrow heads, and it will be noticed that they traverse a considerable distance before escaping to the uptake.

Remarks.—The larger downcomer tubes, as fitted in the

British ships, are probably preferable to the smaller tubes and to the horizontal connecting pipes, because they are less liable to become choked with scale, and are more easily cleared from it. In boilers of the nearly vertical tube type the scale is seldom deposited in the generating tubes, and generally finds its way into the bottoms of the water-chambers, through the downcomers, where it is swept into the parts remote from the more vigorous circulation.

The Thornycroft-Schulz boilers are a considerable improvement on the original *Daring* type, from which they have gradually been developed. The tubes have only a moderate, and not inconveniently acute, curvature. They are not less accessible for removal and renewal than in other types, except the White-Forster, and can be cleaned both externally and internally by any efficient manual appliances. The number of changes in direction, the meeting and mingling of the gases, and the long distance they have to travel before escaping to the uptake, all favour economy in generation of steam; but a greater efficiency is obtained, as shown by results of trials, with boilers in which the gases traverse the passages at right angles to the tubes.

Fig. 48 shows the arrangement of the boiler mountings, and may be considered typical of boilers arranged on the general plan described in this chapter.

Thornycroft Improved Boiler.—Fig. 49A shows the latest form of Thornycroft boiler, as fitted in H.M.S. *Tartar* which attained 35·363 knots on the official six hours' trial. The fuel used is heavy oil, and there are six boilers working at 220 lb. pressure per square inch. Each boiler has 5300 square feet of heating surface; the tubes next the fire are $1\frac{3}{8}$ inches in diameter, and 10 L.S.G. (= 0·128 inch) thick, and the remainder are $1\frac{1}{8}$ inches in diameter, and 12 L.S.G. (= 0·104 inch) thick. The lower part of the upper or steam drum is well protected from direct heat from the fire; this protection appears to be necessary because fatigue of the tube plate has in at least one instance developed into a crack in the tube plate. The general construction of the boiler is similar to that of others of the same type, except for the general curvature of the tubes and the absence of either grips, as shown in Fig. 46, or of baffles in the uptake, as shown in Fig. 44.

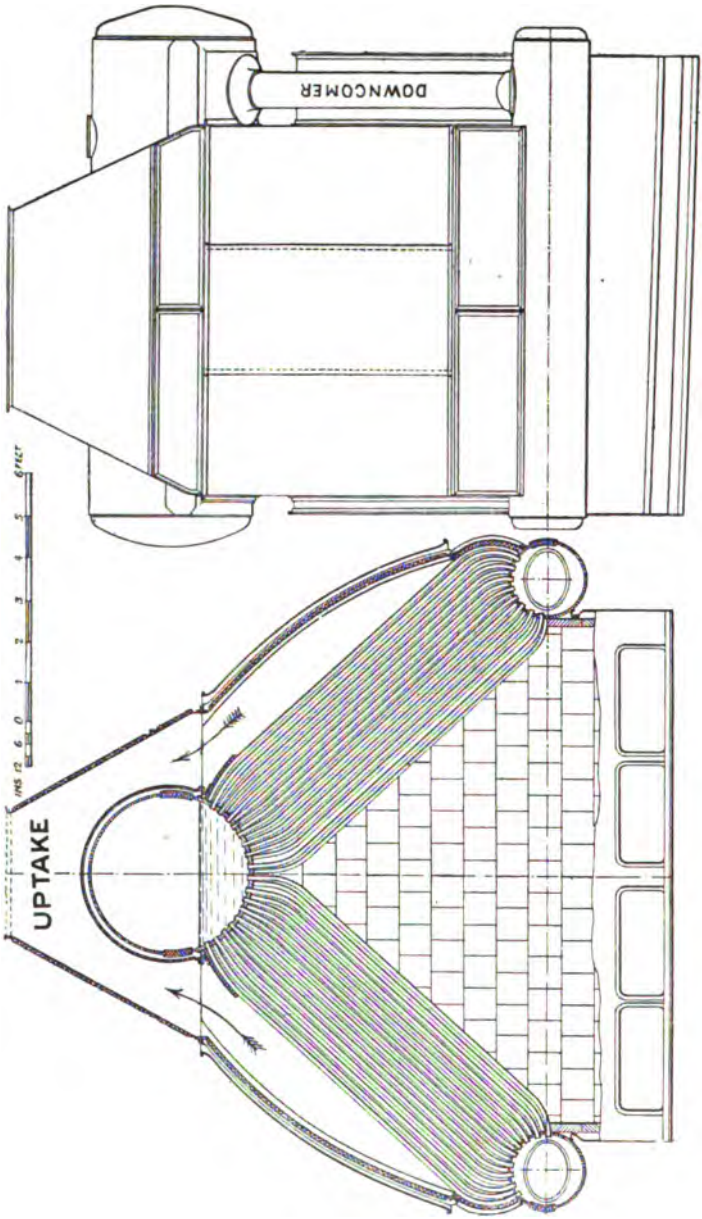


FIG. 49A.—Thornycroft *Tartar* Boiler.

LAIRD-NORMAND BOILER

FIGS. 50 and 51.

Typical Features.—Nearly the whole of tubes deliver their contents below the working water-level. The curvature of the tubes is moderate, and is varied at the back end. Water-walls are made in each wing, completed by baffles. Brickwork is used to screen off the tubes at the back end, for about one-seventh of the total length of the boiler, from the direct action of the fire. Air for combustion is admitted through the casing, and then through small holes in the brick walls at the front and back ends of the fire. The air admitted below the fire at the back end can be regulated by levers worked from the stoking platform, as in the Thornycroft and other boilers.

Circulation.—The water-chambers take their supply of water from the steam chest through large downcomers fitted to the boiler at the stokehold end. A cross connecting pipe connects up the two water-chambers at the back end, and tends to maintain a common level in the two chambers.

Path of the Gases.—The gases rising from the fire pass among the tubes not screened off (as shown in figure), and then, moving towards the back end where they come in contact with the remaining tubes, pass into the uptake at the back end of boiler.

Remarks.—The curvature of the tubes, as in other boilers, is less acute than in the earlier types, and no difficulty should be found in properly cleaning them both inside and outside by manual appliances. Large portable doors are fitted for access to the tube exteriors from the stokehold, for cleaning and sweeping the exteriors and clearing out dirt and ashes. The size of the steam chest is large, and the water area is ample for the power. Three or four long stays are used to stay the front and back ends of the steam chest together. The Bevis feed regulator is generally fitted to these boilers.

VICKERS-NORMAND BOILER

Vickers-Normand Boiler.—The arrangement of this boiler is generally very similar to that of the Laird and other Normand types. The principal difference consists in the path of the gases, which rising from the fire pass through the tube passages at the

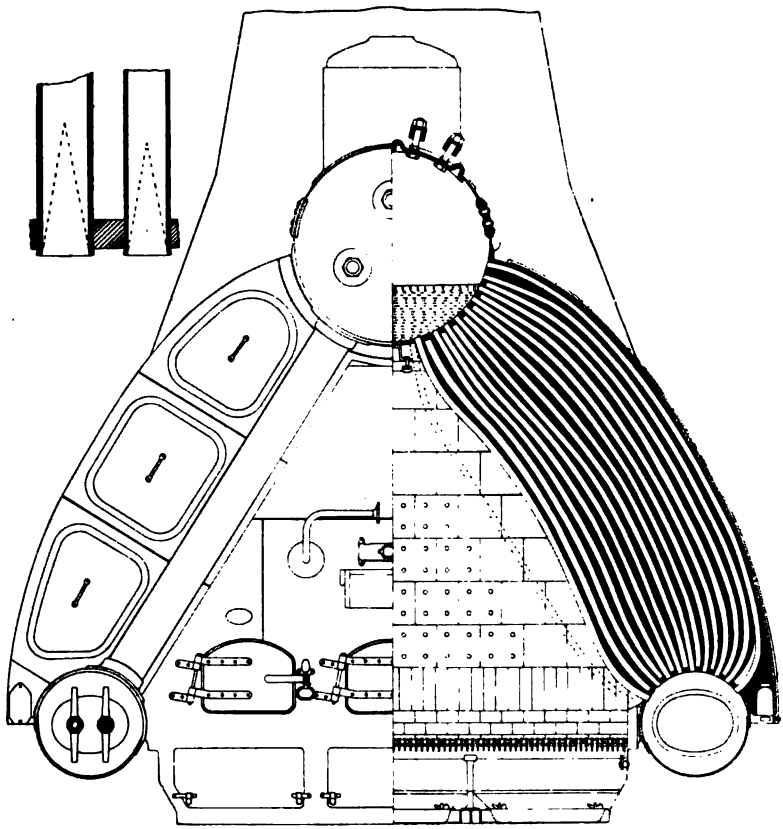


FIG. 50.

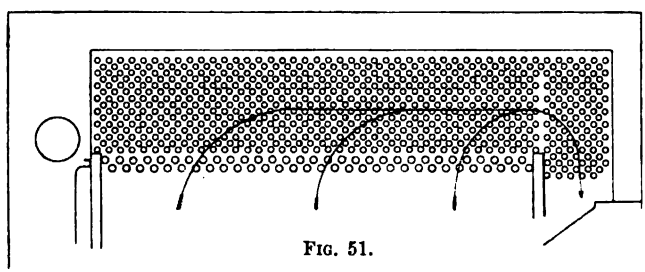


FIG. 51.

FIGS. 50 and 51.—Normand-Laird Boiler.

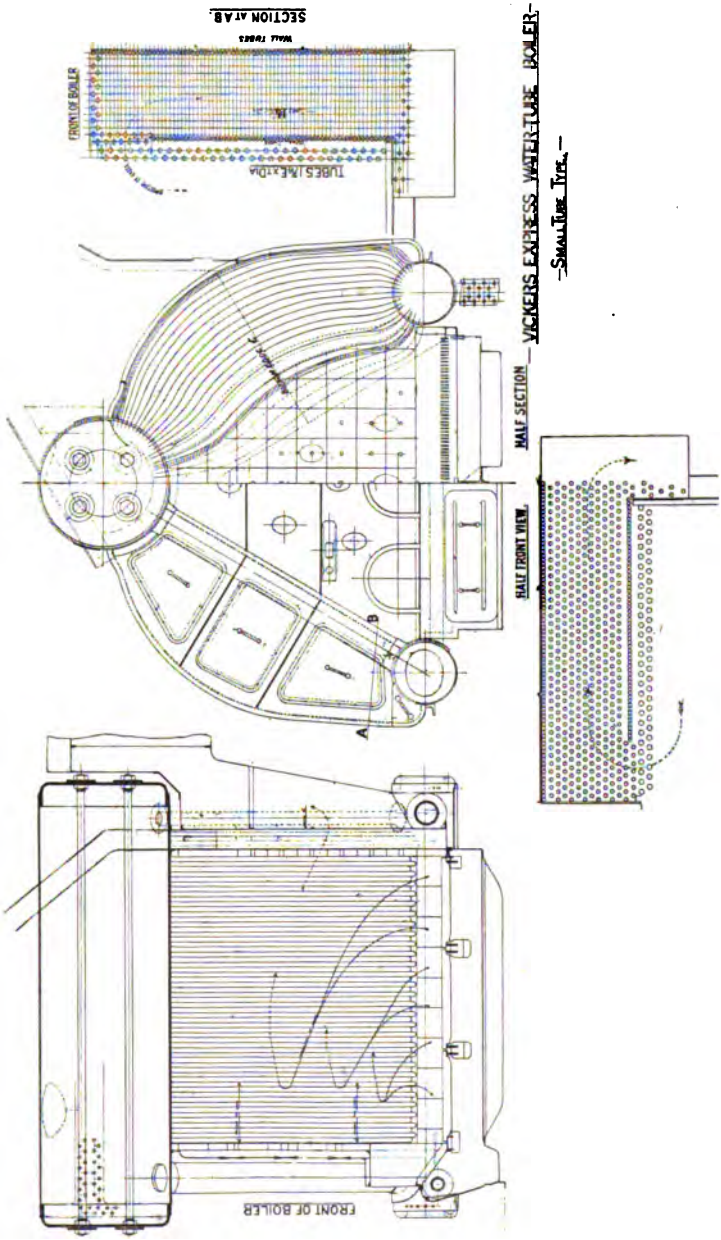


FIG. 52.—Vickers-Normand Boiler.

front end, and travelling nearly horizontally from front to back enter the smoke box at the back on their way to the uptake. The general arrangement is clearly shown in Fig. 52.

Reed-Palmer Boiler

Fig. 53.

Typical Features.—There are two water-chambers below and one steam chest above, joined by bent generating tubes. The

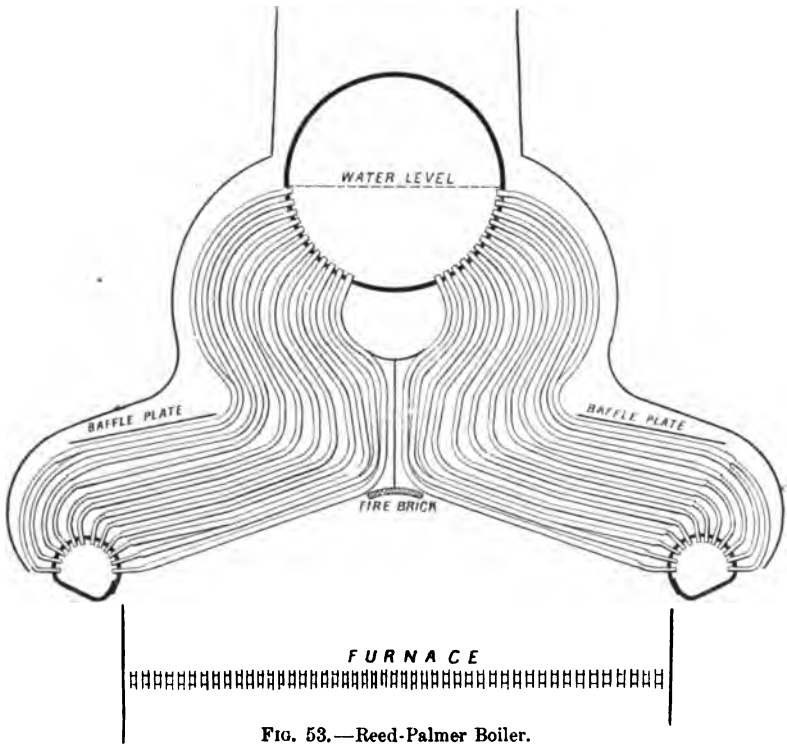


Fig. 53.—Reed-Palmer Boiler.

majority of the tubes deliver their contents below the normal working level. In the earliest types each tube was secured to the tube plate by a gun-metal sleeve, screwed on the end of the tube, which made a metal-to-metal joint with a spherical socketed hole in the tube plate. The sleeve was drawn tightly into place by a gun-metal nut fitted on the water side of the tube plate. The

lower end of the tube was generally reduced in diameter; but as there is some danger from the settlement of scale in the contracted area, this practice is now abandoned. The curvature is acute; but some modification has been made in the more recent designs, so that a special cleaning brush can be drawn through each tube.

Path of the Furnace Gases.—An almost free passage is allowed for the gases from the fire to the uptakes on each side of the steam collector, except where the baffles are placed. Partial water-walls are formed in wake of the baffles as far as possible.

Remarks.—A small steam dome is generally fitted over the top drum and in the uptake. This boiler is similar to some of the Du Temple type in the fixing of the tubes, but in the Reed type the curvature of the tubes is less acute. The grate, as in some other boilers, is fitted at some distance below the water-chambers.

EXTERNAL DIAMETER AND THICKNESS OF TUBES IN VARIOUS BOILERS

Type of Boiler.	Two Rows next the Fire.		Two next Rows		Remainder.		Rows in Uptake.	
	Diam.	Thick.	Diam.	Thick.	Diam.	Thick.	Diam.	Thick.
	in.	in.	in.	in.	in.	in.	in.	in.
Belleville, generator	4·5	0·343	4·5	0·281	4·5	0·192	2·75	0·160 (Economiser)
Belleville	4·0	0·3125	3 1/16	0·25	3 1/16	0·192	2·0	0·144 (Economiser)
Babcock and Wilcox	3 1/16	0·281	3 1/16	0·218	3 1/16	0·160	4·0	0·192 (Top return tubes)
,, ,,	3 1/16	0·312	1 1/16 (Four rows.)	0·144	1 1/16	0·128	4·0	0·192 (Top return tubes)
Dürr	3·5	0·252	3·5	0·212	3·25	0·160	2·75	0·160 (Steam drier)
Niclausse, English types	3·3	0·236	3·3	0·236	3·3	0·160	...	...
Niclausse, French battleship	3·23	0·137	3·23 (Four rows.)	0·137	1·575	0·137	...	...
White-Forster or Yarrow, large tube	1·75	0·1875	...	...	1·75	0·156	...	...
White-Forster or Yarrow, small tube	1 1/8	0·128	...	...	7/8	0·104	...	...
Normand-Laird	1 7/16	0·128	...	...	1 3/16	0·104	...	...
Thornycroft-Schulz, destroyer	1 3/8	0·128	...	...	1 1/8	0·104	3 1/8	0·192 (Downcomers)
White-Forster, torpedo-boat	1 1/4	0·128	...	...	1	0·104	...	...

Tubes.—In the later types of Babcock and Wilcox boilers, the diameter of the tubes, not in immediate contact with the fire, has been decreased to $1\frac{3}{4}$ inches. The inclination of the tubes in the

Babcock boiler is only about 15° to the horizontal; when the inclination is far greater, as in the boilers mentioned in this chapter, this diameter of tube is possibly more than necessarily great. The advantage of small tubes is that a greater heating surface can be obtained, and therefore greater power, on the same weight and space. The disadvantage is that the weight of water carried in the boiler at working height may be so small that automatic feeding is an absolute necessity.

For example, the increase in diameter of each tube, from 1 inch to $1\frac{3}{4}$ inches, increases the water-passage area about $2\frac{1}{2}$ times, while the heating surface of each tube is increased $1\frac{3}{4}$ times; both, for the same power, increase the weight and space. If the same area of heating surface is allowed for a given I.H.P., the weight of water contained in the tubes is increased in the ratio of $2\frac{1}{2}$ to $1\frac{3}{4}$, or 10 to 7, which is equal to about 43 per cent.

When the tubes are only slightly curved, without pockets, and with the axis inclined above 45° with the horizontal for any part of their length, there is very little chance of any permanent deposit under the usually good conditions of working with clean fresh water. Such tubes can be safely made smaller in diameter than tubes with an acute curvature, which makes a part of the tube nearly horizontal.

The *curvature of the tubes* should only be sufficient to prevent distortion when heated, and the curvature necessary for this appears to be about $\frac{1}{4}$ inch per 1 foot of length. Such curvature need not be less in any boiler, and leaves the interior free for cleaning and examination, without any probability of choking. The curved-tube boiler is preferable, and is probably more durable, because the tubes are not then subjected to continual distortion and unnecessary fatigue of material at every time of raising steam or on varying the rate of forcing the fires.

Water-Walls.—In many of the boilers with upright tubes, instead of employing heavy brickwork, walls are formed by the tubes themselves to deflect the furnace gases. Two rows of tubes, differently curved at the ends, so that for some part of their length they block any passage-way between them, form a water-wall. The pitch of the tubes must bear a correct proportion to their diameter to form a wall. If necessary, the openings which are left near the tube plates are backed by plating or baffles, to complete the wall. Water-walls are commonly fitted in boilers of the Thornycroft, Laird, Vickers, Blechynden (old), and Reed types.

Plugging, or Renewing, Defective Tubes.—A defective tube in an upright tube boiler is generally plugged and left until others require renewing, or until it is convenient to renew it. A few tubes less do not affect the rate of evaporation materially, and if properly plugged they are quite safe, as found by practical experience. The plug used is generally a smooth cone made of steel, and inserted in the end of the tube from the inside of the boiler. The pressure inside the boiler keeps the plug in place, and a flange is left on the plug so that it cannot be driven through the tube plate, either by a hammer or by pressure. Of course both ends of a defective tube

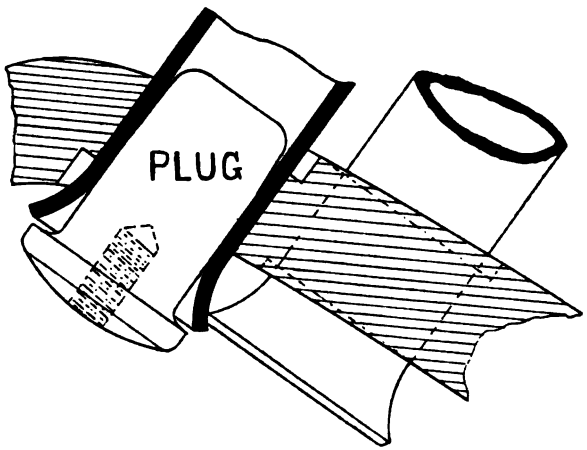


FIG. 54.—Plug for Upright Tube.

are plugged, and when steaming the tube burns away, but parts in contact with the water remain intact.

In many types the renewal of a tube is a matter of considerable difficulty, unless the tube is in the outer rows. Where the tubes are simply expanded into their holes in the tube plate, the ends are generally bell-mouthed, to prevent their accidental withdrawal. The bell-mouthed ends must be cut off before the tube can be withdrawn, and if any tubes are in the way of the withdrawal these must be first removed. A tube once removed is generally unfit for replacing. Special means are adopted for withdrawal and renewal of tubes in the White-Forster, Fleming and Ferguson, Cummins, and Blechynden boilers. The first-mentioned appears the most practical.

Steam Collector.—This is generally constructed of two semi-circular plates, double riveted together longitudinally with either

lap or butt joints. Double butt joints are more usual and are generally specified. The upper plate is of uniform thickness; but the lower one, which forms the tube plate and receives all the upper ends of the tubes, is thicker, except where it is machined down to the thickness of the upper and end plates to form the joints with them. These two plates form a circular shell, which is closed at the ends by two domed or dished plates, flanged as necessary to fit the circular shell. The front plate is riveted to the shell, and presents a rounded face towards the stokehold. The back end is sometimes made with an external joint, for convenience of manufacture and for machine riveting; but the dome should project outwards in all cases.

All the necessary boiler mountings are fitted on the front plate, except the safety valves and air-cock, which are more conveniently fitted on top of the circular shell. A manhole is fitted as convenient, but generally in the front plate and clear of the boiler mountings.

The steam-chest is fitted internally with the necessary zinc protectors and internal steam-pipe. In addition, nearly all water-tube boilers are fitted with a feed regulator and float gear, either partly or wholly outside the chest. Boilers with above-water delivery are also fitted with internal separator plates. (See Figs. 55 and 56.)

Water-Chambers.—The construction of the water drums is very similar to that of the steam-chest. Two plates are used, one thick and one thin, but the shape of the drum is frequently only partly cylindrical, as shown by the figures. In all cases the part forming the tube plate is thicker, and when the plate is flattened the thickness is variable to withstand distortion. A manhole is now usually fitted to each water drum, at the stokehold end, for access to the interior and to the tube ends for cleaning and repair. A drain valve, or blow-down, is generally fitted to each water drum, with which a hose connection can be made for running the boiler water into the reserve tanks or bilges.

The water drums rest on the boiler seatings fitted in connection with the framing of the ship, and when steam is up the generating and downcomer tubes generally support the upper parts of the boiler. Framing in connection with the ship is also fitted to support the steam-chest when cold, and is specially constructed so as not to check the necessary expansion of the boiler, particularly in height. This framing prevents the boiler shifting on its seating when the ship is rolling and pitching.

THORNYCROFT *SPEEDY* TYPE (1893).

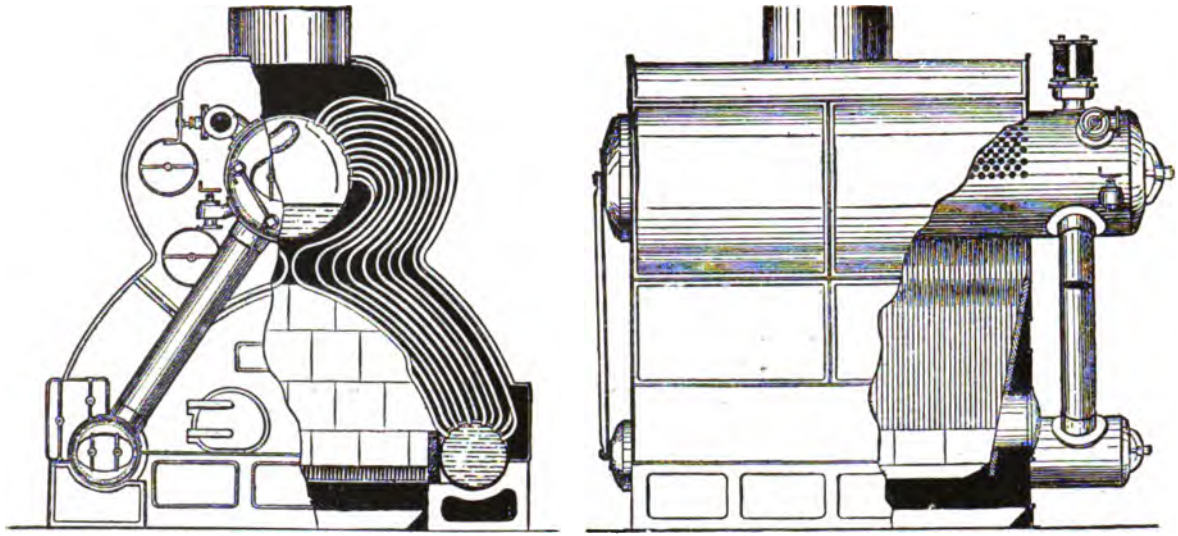


FIG. 55.

Remarks.—First boiler of upright tube type fitted in Navy. Large heating surface and, therefore, economical. Above-water delivery, with screens in steam chest. Partial water-walls next fire and casing. Modifications fitted in several torpedo boats, but not in destroyers.

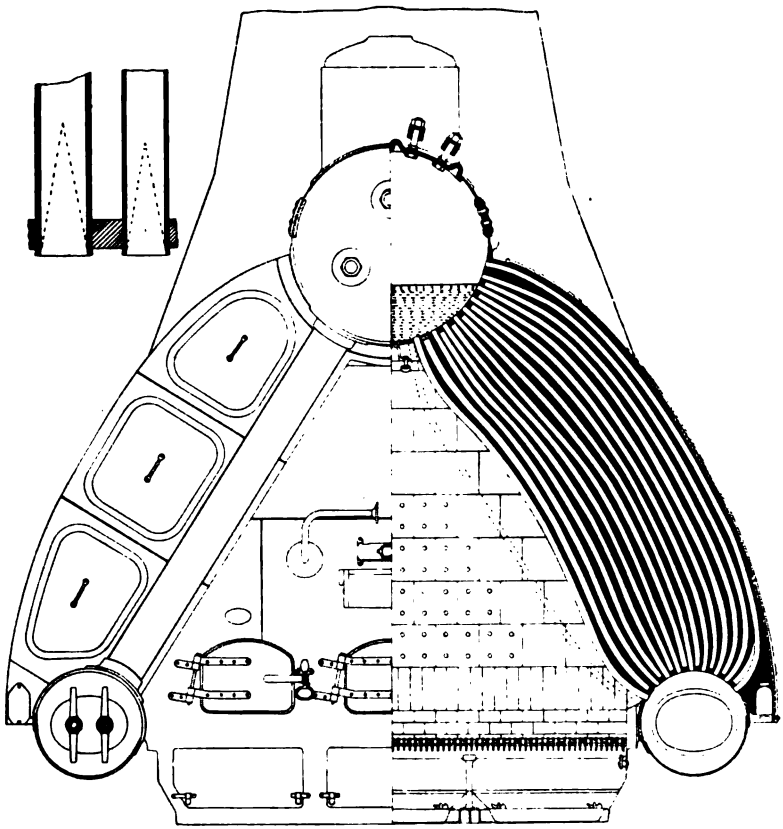


FIG. 50.

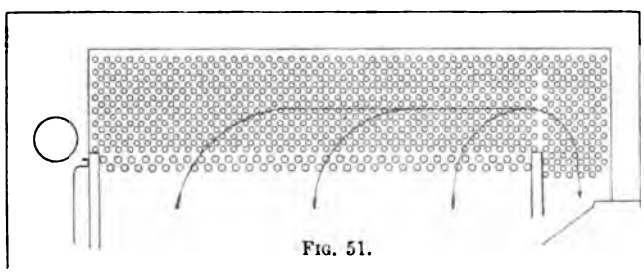


FIG. 51.

FIGS. 50 and 51.—Normand-Laird Boiler.

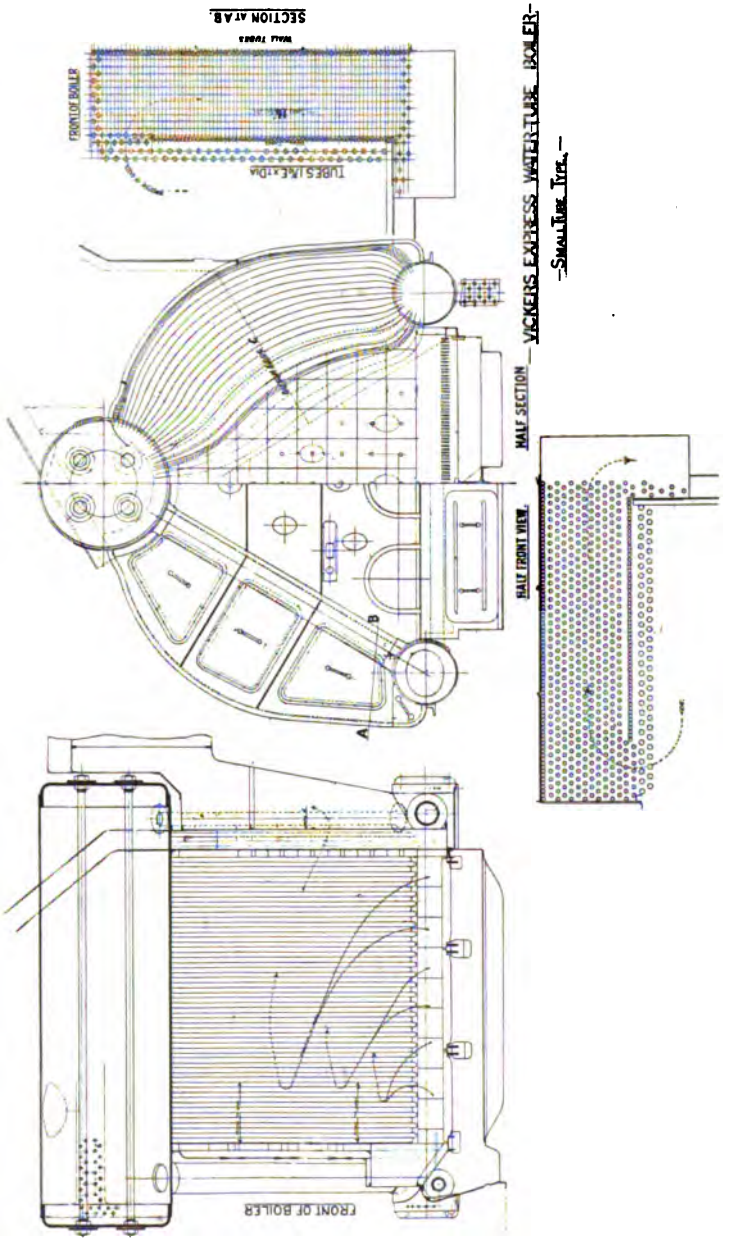
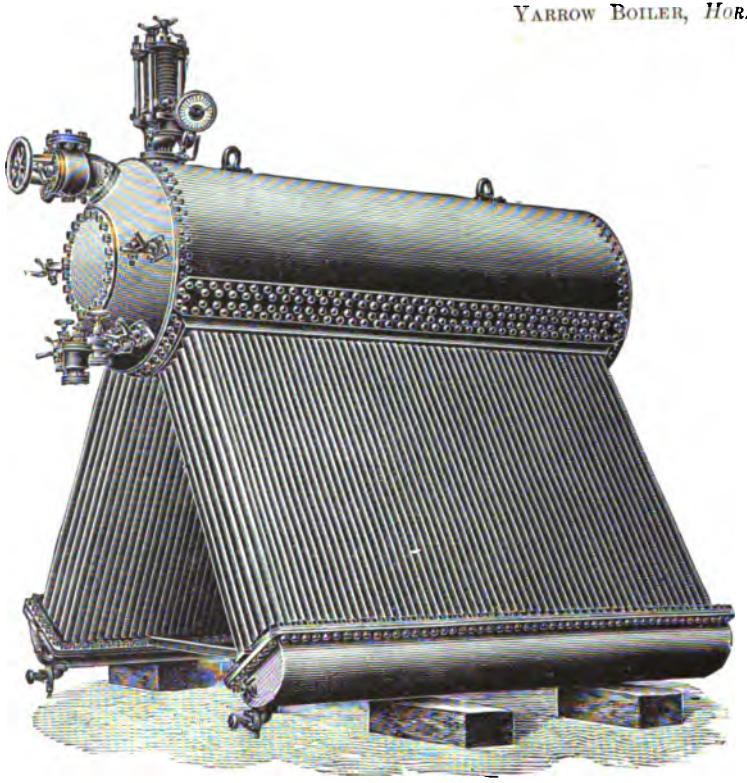


FIG. 52.—Vickers-Normand Boiler.

YARROW BOILER, *HORNET* (1895).



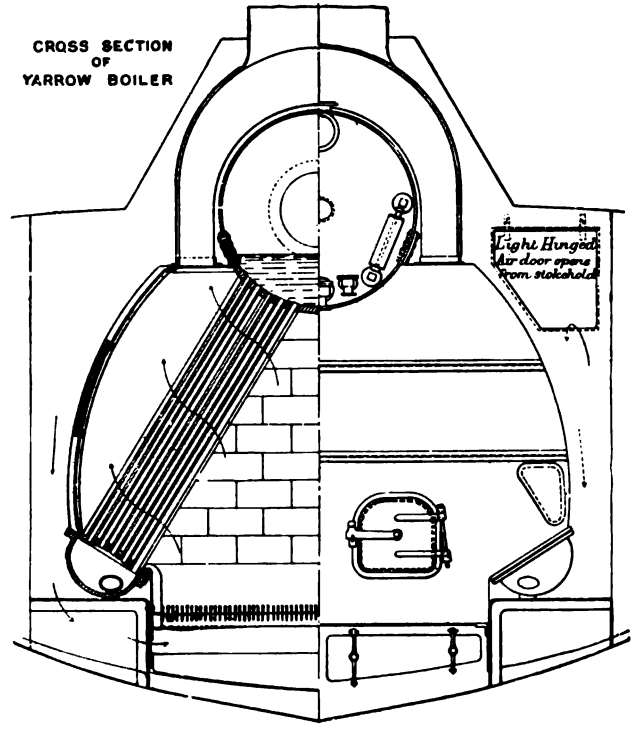


FIG. 58.

Remarks.—Water-chambers with flanged, bolt and nut, joints. Notice course of air-supply to ashpits.

Blechynden Boiler (1895).

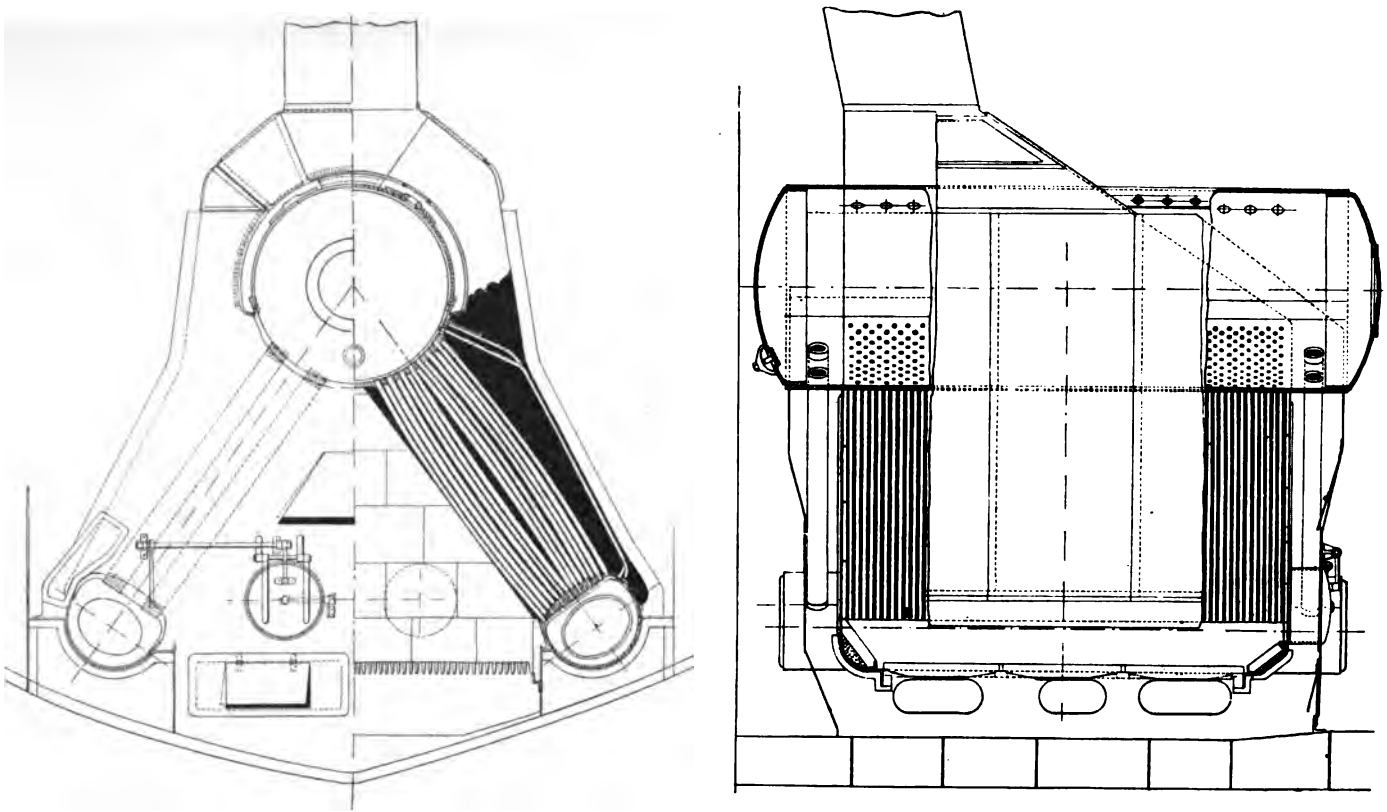


Fig. 59.

Remarks.—Tubes of slight curvature, and can be removed through two rows of holes in top of steam chest; but for practical purposes this method is generally useless. Probably among the best proportioned of early boilers, but copper tubes originally fitted in outer rows spoiled its reputation. Modifications fitted in "P" cruisers gave most contradictory results: some very good and some very bad.

The *Mosher* (American) boiler is a modification of this type, built on the general lines of the White-Forster boiler.

WHITE *CONFLICT* BOILER (1895).

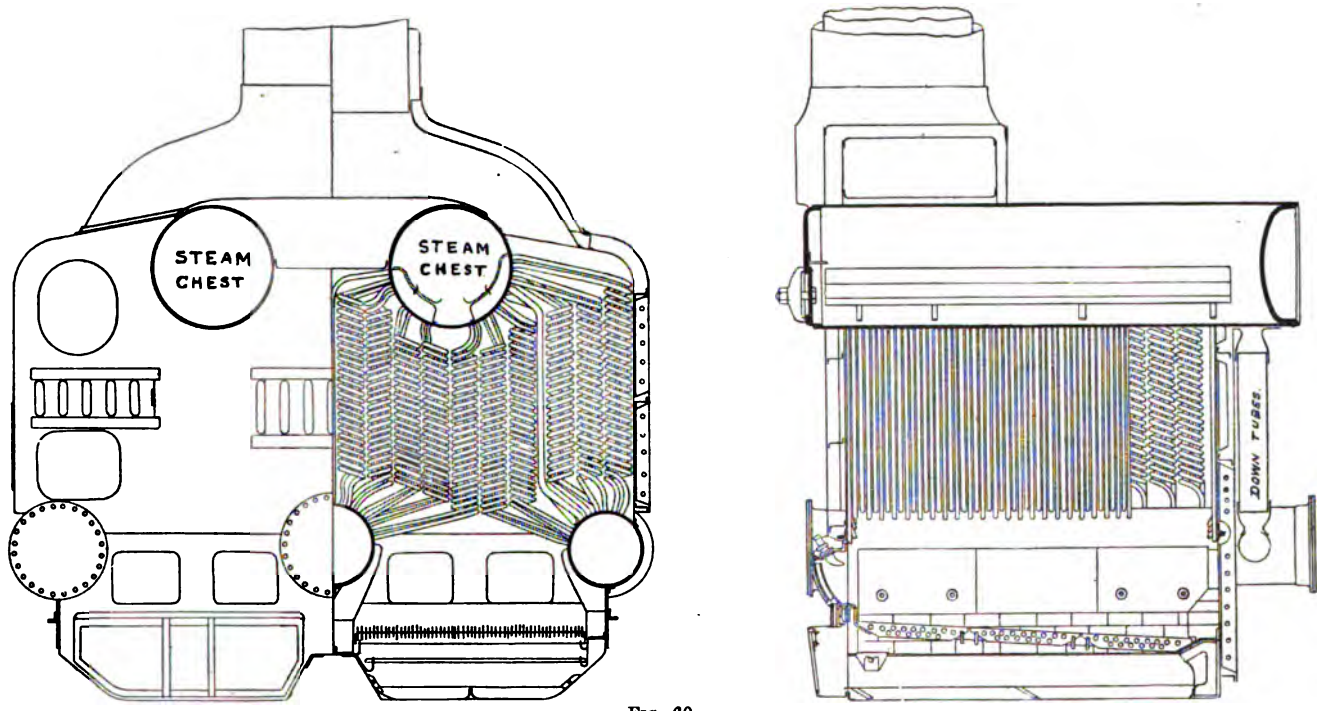


FIG. 60.

Remarks.—Spiral tubes, and tubes with acute curvature, make it difficult to clean either internally or externally. Two upper steam chests produce unnecessary complication.

CHAPTER VIII

WATER-TUBE BOILERS: NEARLY HORIZONTAL TUBE TYPE

Babcock and Wilcox Boiler

Typical Features.—The generating tubes are straight, and inclined at an angle of about 15 degrees with the horizontal; each tube is connected with two headers with wavy sides; and a number of such tubes in connection with a pair of upcast and downcast headers forms an *element*, as shown in Fig. 61. Each downcast header is connected at the top with the steam chest or collector, and at the bottom with the feed collector, which are common to all the elements, 16 to 20 in number, of a single boiler. Each upcast header is connected at the top with the steam chest through a pair of top return tubes.

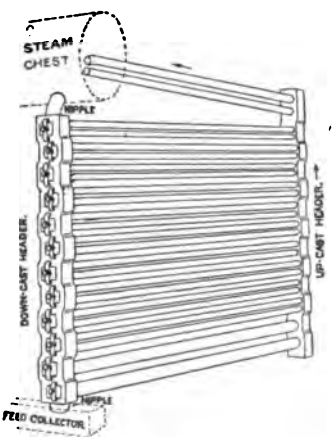


FIG. 61.—Babcock and Wilcox Element.

The tube ends and nipple connections at the top and bottom ends of the downcast headers are made steam-tight in their holes by rolling and expanding them in the usual way. There are two principal designs for naval purposes: one with tubes of $3\frac{3}{16}$ inches diameter throughout, called the medium tube design; and the other containing $3\frac{5}{16}$ inches diameter tubes for the two rows next the fire (as shown in Fig. 61), and $1\frac{3}{16}$ ins. diameter tubes for the remaining rows, called the small tube design. The small tubes allow some increase in heating surface (and power) for the weight,

and for this reason have been adopted for recent naval vessels. As shown in Fig. 61, the sight holes in the headers, for fitting, cleaning, and searching the tubes (to ascertain a clear passage), are square; and for the smaller tubes, one sight hole admits 4 tubes. This arrangement has also been adopted for recent naval vessels.

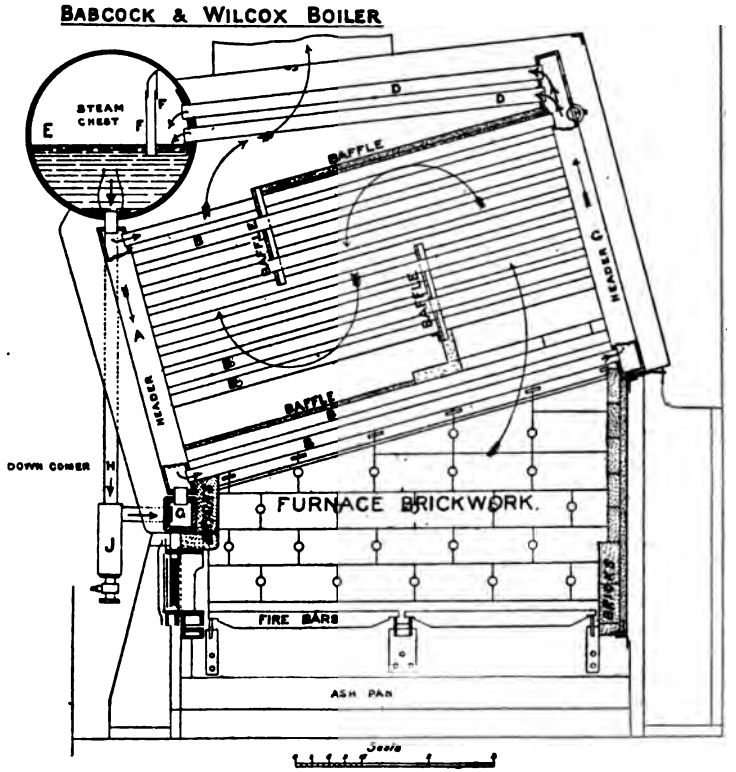


FIG. 62.—Circulation in Babcock Boiler.

Circulation.—At working height the water level is below the horizontal diameter of the steam collector, and the whole of the generating tubes and all internal parts of the headers and feed collectors below this level are therefore submerged, leaving the top of the uptake headers and the top return tubes above the normal working level. The circulation is generally in accord with that described for Fig. 17, but Fig. 62 shows the boiler in fuller detail.

The feed water is pumped into the steam collector E, and is

generally automatically controlled when high rates of evaporation are intended. From the steam collector the water passes down the downtake headers A, and, supplying the generator tubes B, replaces the water evaporated or circulated away into the uptake headers C. The inclination of the tubes sets up a circulation towards their upper ends, and so to the uptake headers; there the steam tries to rise to the top. From the top of the uptake headers, the steam and some water, impelled by the circulation caused by convection, pass through the top return tubes D into the steam collector E. The top return tubes deliver their contents above the working level in a nearly horizontal direction, but slightly downward; the steam and water impinge on dash-plates F, the steam rises to the upper part, while the water mingles with the feed and other water in the bottom of the steam collector, where it begins a fresh circuit.

The bottom ends of the downtake headers are connected together by a horizontal tube G of square section, called by the makers the *mud-drum*, but which is actually a feed leveller and collector. For naval purposes extra *downcomers* H with *sediment* boxes J are fitted to supply this drum with water from the steam collector independently of the downtake headers. Any roll of the ship above 15°, the inclination of the generating tubes, tends to check the circulation, and consequently these tubes are preferably placed in line with the keel, where they are not so much affected by rolling. In this case (always fitted for naval purposes), when the ship rolls the water is prevented by wash-plates from rushing to one end of the steam collector, and thus leaving the tubes at the other end temporarily denuded of water. The extra downcomers ensure some supply to these tubes through the feed collector.

The *wash-plates* are flat plates fitted across the lower part of the steam collector, with their top edge just above the working water level.

Path of the Furnace Gases.—The gases rise from the fire and pass through the passages between the generating tubes, which are in vertical staggered rows, to the space above; thence they pass to the uptake leading to the funnel, between two other staggered rows of tubes, conveying the upcast steam and water from the headers into the steam collector.

When the small-tube design is used, the baffling among the tubes is occasionally omitted; but baffling tends to produce an increased efficiency by retarding the flow of the hot gases to the

funnel, and is fitted in recent warships similarly to the medium-

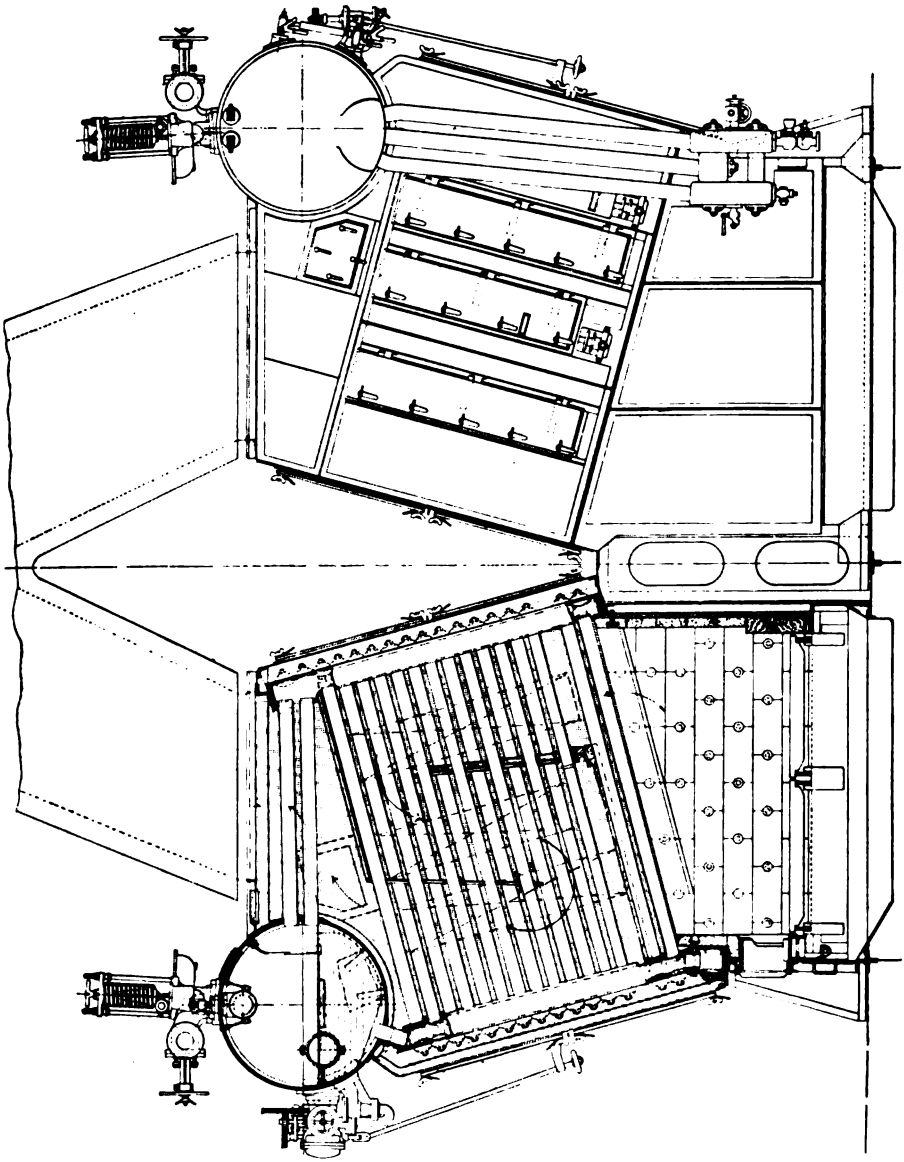


FIG. 62A.—Babcock and Wilcox (medium tube) Boiler.

tube design, indicated in Fig. 62A. The first baffle is fitted above the second row of tubes, and extends about two-thirds of their

length from the front end, then rises parallel to the headers about half-way among the upper rows. The gases are deflected downwards by a second baffle, fitted on top of the top rows, and extending from the back end about two-thirds of their length. They then turn upwards, after passing below an extension of the second baffle, which projects downwards and blocks about one-third the passage-way. Thence the gases pass upwards between the rows of top return tubes, and so to the uptake leading to the funnel.

The path of the furnace gases, as described, embodies an efficient principle. The gases surround, when hottest, the parts of the generating tubes and headers which contain water and steam of the highest temperature, and after being cooled during this process they are conducted away towards the cooler ends of the generating tubes and downtake headers. The top return tubes are, however, in a comparatively cool place; but as the temperature is still probably from 200° to 300° in excess of the steam and water inside the tubes, some heating is obtained from the escaping gases. A feed heater placed in this position is more efficient, and has been fitted with this type of boiler in some installations.

Tubes.—In all cases the tubes are expanded by tube rollers into the headers to obtain pressure tightness, and in addition the ends are bell-mouthed, to resist the tendency to withdraw from their holes. (See "End Pressure," or "Axial Tension," p. 109, etc.)

When the tubes are $3\frac{3}{16}$ inches diameter throughout, or, as called by the makers, "the medium-sized tube design," the thickness of the two rows of tubes next the fire is $\frac{9}{32}$ (0·28) inch, of the third and fourth rows $\frac{7}{32}$ (0·22) inch, and the remainder 0·16 inch thick. The thickness of the top return tubes is 0·192 inch for all designs.

For the small-tube design, shown in Fig. 63, the two rows of tubes next the fire are $3\frac{5}{16}$ inches diameter and $\frac{5}{16}$ inch thick, swelled at the stokehold end to a diameter of 4 inches for a length of about 3 inches. The next four rows are $1\frac{3}{16}$ inches diameter and 0·144 inch thick, and the remaining tubes are of the same diameter, but 0·128 inch thick. All the small tubes are swelled at the stokehold (the withdrawal) end to a diameter of 2 inches for a length of 2 inches.

Steam Collector.—The steam collector is a circular structure placed across the top of the downtake headers, next the stoking platform. The ends of the steam chest are closed by circular

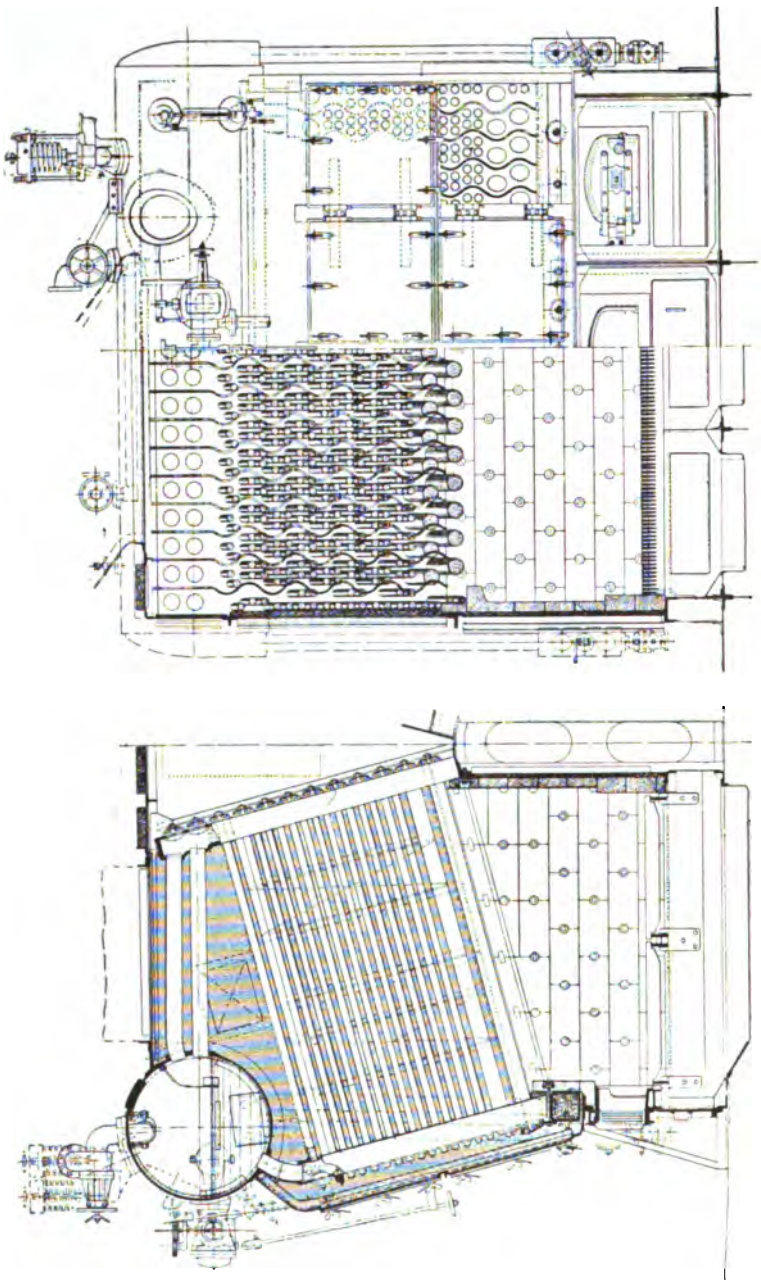


Fig. 63.—Babcock and Wilcox Boiler (small-tube type).

dished plates, and the independent downcomers project downwards, near each end, as shown in Fig. 62A. The collector is strengthened by plating, where the top return tubes and the circulation nipples are connected with it.

All the boiler mountings are attached to the steam collector, with the exception of the blow-down cocks, which are connected with the bottom of each sediment box. If independent downcomers and sediment chambers are not fitted, then the blow-down is fitted to the mud-drum directly. The position of the mountings is indicated in Fig. 83 (Chap. IX.), and the casings in Fig. 104.

Nipple Connections of Headers.—The connections of the headers with the steam collector and with the mud-drum are made through short lengths of tube about 4 inches in diameter, and expanded into the holes to obtain pressure-tightness, as shown in the figure. The ends are also bell-mouthed to resist the end pressure, or axial tension.

Cleaning Doors and Closures.—Holes are made in the forged steel headers for fitting the tubes in place. For the larger sized tubes and connecting nipples, *oval* holes were originally made instead of square, and closed by internal oval doors or closures. The inside face of the header around each hole is machined to a true surface, and on this the joint of each door is made with a jointless asbestos ring. The smaller holes are closed either by a coned plug, with its apex projecting outwards, which is drawn up to its seating by a faced nut, making another joint on the outside, or by a rectangular door opposite the ends of a group of 4 tubes, as shown in Fig. 61. Provision is made for access to both ends of the tubes by smoke-box doors of the usual pattern. Generally, corrugated plating is used for all parts of the casing and smoke-box doors.

BELLEVILLE BOILER

Typical Features.—The excessive coal consumption of the early Belleville boilers led to the adoption of the feed-heater, commonly known as the *economiser* type, in which the number of tubes is reduced to 14 or 16 in each *generator* element, and the weight and heating surface thus saved is transferred to a smaller set of tubes and elements fitted above the generating tubes, and called the economiser. The water, on its way to the generator, passes through the economiser tubes, and as these latter are situated

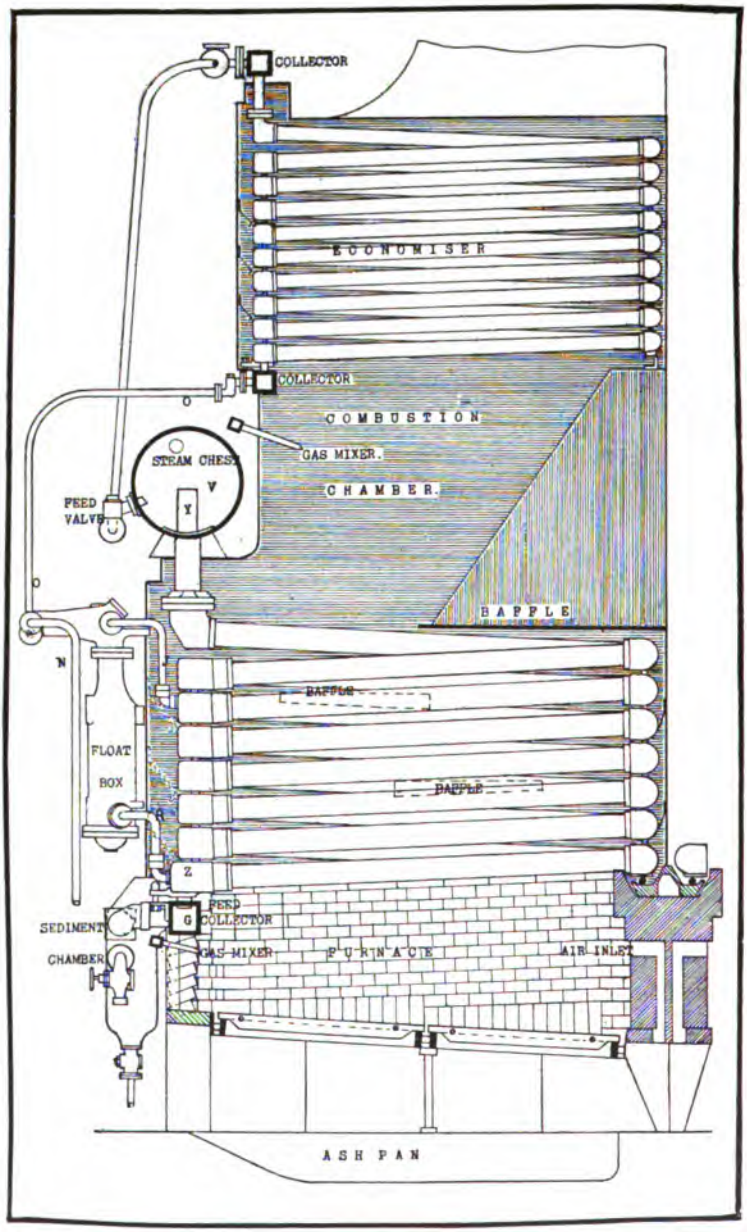


FIG. 64.—Belleville Boiler and Economiser.

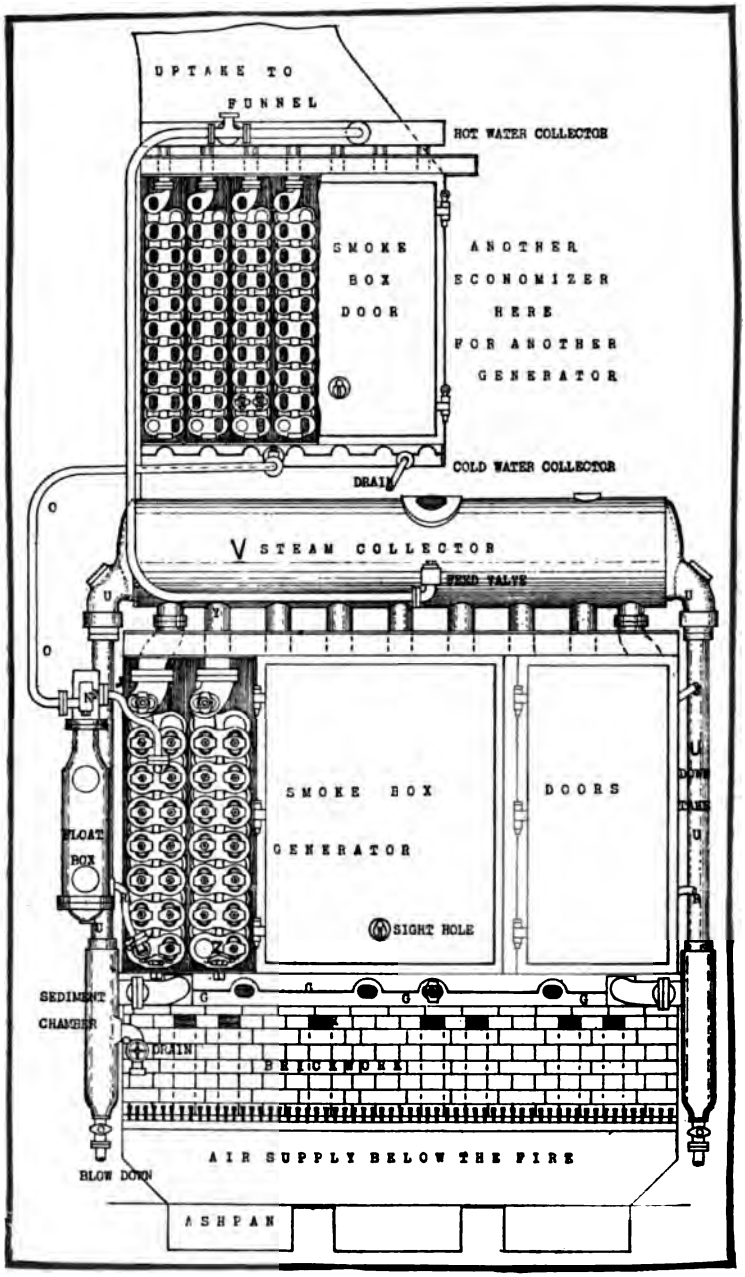


FIG. 65.—Belleville Boiler and Economiser.

in the base of the funnel, a certain quantity of heat, otherwise lost, is transferred to the feed water from the escaping gases.

The economiser is fitted with the Belleville boiler as shown in Figs. 64 and 65, and the path of the furnace gases is also clearly shown.

Commencing with the lowest junction box, the generator tubes are inclined upwards, at an angle of from $2\frac{1}{4}°$ to $2\frac{3}{4}°$ to the horizontal, by raising opposite ends alternately. The junctions are formed at the ends of each two tubes, one of which is inclined downwards to the junction box below, and the other upwards to the junction box above. The lowest tube in each element is connected with the *feed collector G*, and the uppermost with the *steam chest V*, by a pipe, *Y*.

Course of the Feed Water.—The feed water enters the automatic feed regulator from the feed pipe *N* under a pressure of nearly double that of the working pressure in the boiler. From the feed regulator it passes through a pipe, *O*, and a non-return valve, into a horizontal collecting pipe called the *cold* water collector at the bottom of the economiser elements, and is distributed among them. The water pumped through the economiser is heated during its passage, and escapes through another horizontal collecting pipe, called the *hot* water collector, connecting up the top ends of the economiser elements; thence it passes through a pipe and non-return feed valve directly into the steam chest of the generator, where it is deflected downward. From the steam chest it passes, with the other water in the lower part of the steam chest, into the *downcomers U*, which consist of two large vertical pipes in connection with the ends of the steam chest. At the bottom of each downcomer the water passes through a non-return valve into the *sediment* chamber, where any dirt or grease should be deposited, thence through a short elbow pipe into the feed collector, *G*, whence it is distributed to the generator elements. The water as it is heated again rises, mixed with the steam which is generated in the tubes. This causes sufficient circulation to eject the steam, mixed with a fair amount of water, through the pipe, *Y*, fitted at the top of each element into the steam chest. Above the discharge pipes a mushroom top is fitted, so that the mixture is deflected downwards; the water falls to the bottom of the chamber, and the lighter steam escapes upwards into the upper part of the steam chest. The water thus separated out from the steam mixes with the incoming feed water, and recommences a fresh circuit of circulation. The

steam passes through small holes in the internal steam pipe into its interior, whence it is led away by the steam valve and pipe to the engine, or it escapes through the safety valves.

The economiser is entirely filled with water under a pressure considerably above that in the generator. No steam is formed in the economiser because the boiling-point due to the pressure is never reached in practice.

Junction Box.—In Fig. 66 a horizontal section and front elevation are shown of the ordinary junction box at the front end of the tubes. In the Belleville boiler it will be noticed that the usual practice of expanding the tube ends with rollers to make the joints steam-tight is not carried out, but screwed joints are employed. The junction box varies slightly in form and shape, as made by different firms, but the principle is the same in them all. There are two spigot pieces cast on it, into which the tubes, or ferrules for connecting the tubes, are screwed, with a slightly different pitch of thread, 10 to the inch on the tube or ferrule, and 10·16 to the inch on the box and sleeve piece. The slight difference of pitch tends to secure a tight joint. The sleeve piece is jointed with *graphite* cement and a check nut at the outer end. Belleville graphite packing consists of about 40 parts of tin, 46 parts of lead, 4 parts of mineral oil, and 10 parts of graphite or powdered black lead. A blunt chisel or set is used for hardening up the nuts and sleeve pieces where space does not permit of gas tongs being used.

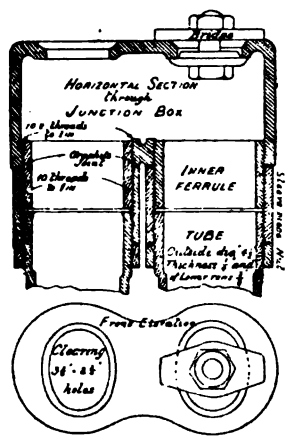


FIG. 66.—Junction Box (front).

Opposite the end of each tube a small oval door, or *closure*, is fitted, for clearing and sighting the interior. The door is secured by a stud, made in one piece with it, and an external bridge- or dog-stay and nut. This closure is jointed with a jointless asbestos wire-woven ring, not more than $\frac{1}{8}$ inch in thickness.

The screwed ferrule[1] and sleeve piece are necessary to connect

[1] The screwed joint with ferrule, spigot, nut, and packing, was previously used and patented in 1831 by Sir Charles Dance for a water tube boiler for motor wagons, and is curiously similar to that fitted in Belleville boilers. (See *Motor Vehicles and Motors*, vol. i., by Worby Beaumont.)

up every second tube, because after one tube is screwed into a front and back junction box, it is impossible to screw on another tube and junction box without some such connection. The junction box at the back end is very similar as regards the securing of the tubes, but the sleeve piece and ferrule are omitted, as each tube is screwed directly into the junction box. No clearing holes are fitted at the back end. Lugs are cast on top and bottom of each junction box, to support the weight of those above it. They are side slotted from front to back, so that each junction box can move in the direction of expansion of the tubes independently of those next it. The lowest front box is supported on the feed collecting tube, which runs right across the front of the boiler; the lowest box at the back end is supported on loose rollers, moving on a part of the casing specially prepared for the purpose.

The uppermost junction box is of different form and shape, to provide for the connection with the steam chest; a square flange is formed at its upper part, and is secured by four bolts and nuts to a similar flange on the discharge pipe. The upper flange is spigoted into a recess in the lower flange, with a jointless asbestos ring between the two butting faces inside the recess.

Bottom Junction Box.—The manner in which the lowermost junction box is connected with the feed collector is shown in Fig 67. It is very similar in form to the ordinary front box, but the second clearing door is omitted, as only one tube is connected with it. Below the tube connection and on the under face a coned hole is made, which rests over a steel coned-nipple, which is screwed (with a differential thread) tightly into the top of the feed collecting tube; this screw is slightly coned for the purpose. To make a tight joint between the nipple and the coned hole, a *nickel ferrule* is used, and the general practice is to use two ferrules to each nipple, as it makes a tighter joint. A hole about $1\frac{11}{16}$ ins. in diameter is drilled through the nipple, and this hole forms the

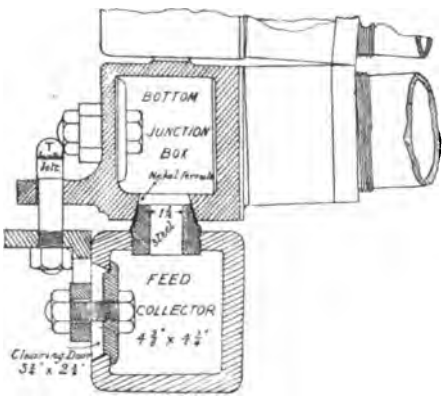


FIG. 67.—Bottom Junction Box.

water connection between the element and the feed collector. The junction box is secured down on the nipple by the *anchor bolt*, which is T-headed above a lug cast on the front of the box, and is nutted below an angle steel riveted to the front face of the feed-collector. Sight holes and closures are fitted to the collector tube, four or five holes to each boiler as convenient. At each end the collector is connected with the sediment chamber by a cast-steel elbow pipe. When there are two downcomers, as usually fitted to boilers of larger size, there is a possibility of the water circulating unevenly, or of a reversal of current. To prevent this, a non-return flap valve is fitted in the upper part of each sediment box at its junction with the downcomer; this valve is shown in Fig. 69.

Steam Chest or Collector.—In Fig. 68 *Y* is the junction of an element with the steam chest *V*. When the boiler is at work, steam and water are ejected upwards from the generating tubes or elements, and impinge on a dash-plate, until finally the steam passes through small holes (one of $\frac{1}{16}$ inch for each element) into a steam-tight chamber.

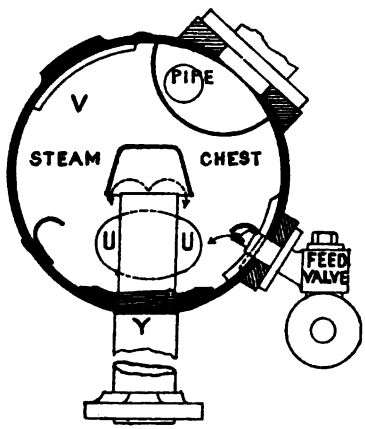


FIG. 68.—Upper Junction Box and Steam Drum.

Where the junction of the element to the steam chest is made, a doubling plate is riveted inside the steam chest to give it sufficient local strength. Near the bottom of the steam chest, towards the front, two or more oval doors were sometimes fitted, which could be easily removed for cleaning and examining the interior of the drum. At each end, generally, a manhole door is now fitted, so that by removing it from the steam chest an examination of the interior parts can be made. The connection *U* for the downcomer is generally made on the end plate, below the manhole door. The connecting pipe *Y* is screwed into the steam chest, and the joint is made tight by fitting a differential thread, as used for the tube connections, and by making the pipe on a slight cone. The pipe projects into the steam chest about one-third the diameter of the latter, and thus as the water level is always below its top, prevents any tendency to reverse the direction of circulation,

and tends to produce a continuous circulation in the desired direction. The feed water enters the steam chest near its lowest part, and a short internal pipe directs the feed water downwards.

Sediment Chamber.—This is simply a bottle-shaped enlargement, as shown in Fig 69 on the bottom of each downcomer. It acts as a quiet place, in which there is little circulation or movement of the water, so that generally any solid matter that may be brought down to it remains, and can be removed by opening the blow-down, which, with an internal pipe rising about 5 inches from the bottom, is attached to its lowest point. It is found in practice that nearly all the sediment is so deposited, if the density is practically *nil*; but if the density is evident on the hydrometer, care should be taken to keep the sediment box as free as possible from accumulations of solid matter by frequent blowing out. A small door is fitted on the box, so that the non-return valve can be taken out and cleaned and kept in working order. Any accumulation of matter affects its efficient working. A drain valve is fitted on one sediment box of each boiler for emptying the boiler when required.

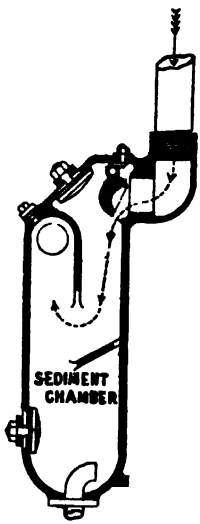


FIG. 69.—Sediment Chamber.

Automatic Feed Regulator.—The proper regulation of the water level is the most important factor in the satisfactory working of a Belleville boiler, and generally of all water-tube boilers. At full power the weight of water contained at working level in an ordinary tank boiler must be renewed in about $2\frac{1}{2}$ hours, while in the Belleville it must be renewed every 14 minutes, consequently the satisfactory maintenance of a constant working water-level is almost impossible by hand regulation alone. The difficulty of hand regulation is increased by the greater number of boilers, and an automatic feed regulator is almost an absolute necessity.

Standing out a little from the front of the boiler, and generally on the right hand when looking towards it, is the automatic feed regulator, shown in Figs. 64, 65, and 70. There is a cast-steel chamber, in which a float, *M*, can move either up or down, within the limits of the chamber. The upper part is connected with the

second or third junction box from the top of the element, through the pipe J; the bottom of the chamber is connected through the pipe R, with, generally, the bottom box Z. Through these two connecting pipes, R and J, a water-level is obtained in the float chamber corresponding with the head of water in the element with which it is connected. Any alteration in the head of water in the element operates on the float M. At the bottom the float is fitted

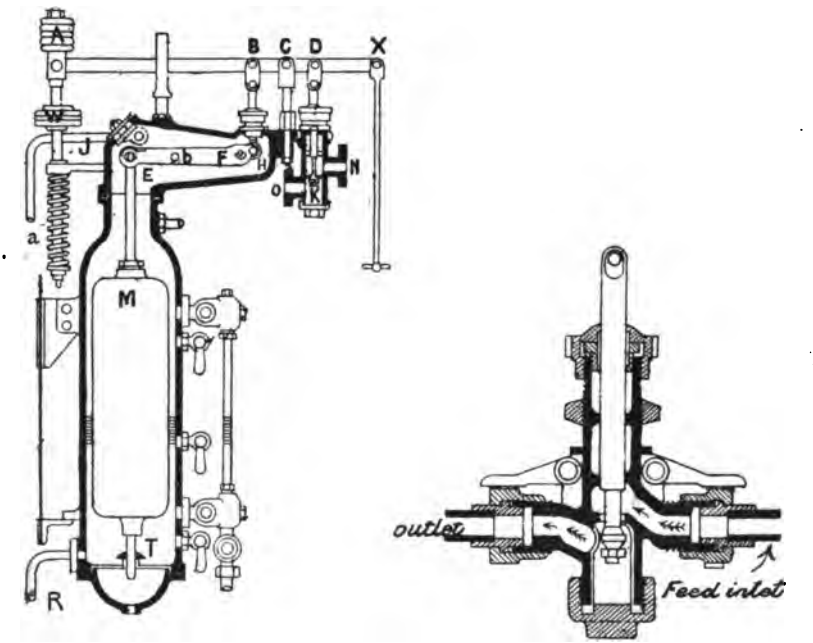


FIG. 70.—Automatic Feed Regulator (Belleville).

with a guide rod, working in a bracket, T. At the top the float is connected by a vertical rod, EM, with the lever EFH, which has its fulcrum at F; consequently any movement of the float operates on the rod BH, and by it on another lever, $ABCDX$, exterior to the chamber. This lever has its fulcrum at C. A rod, DK, works the feed valve K, which opens downwards. At A a small adjusting spring, a, and weights, W and A, are placed; these are used for any small adjustment of the proper equilibrium of the float and levers. By adding weights, it is seen that the float is allowed to rise more easily; while by removing them there is a greater

tendency for it to fall. Any slight difference required in the normal working level can be kept constant by adjusting the weights. The water-level tends to rise when the combustion is more active with an increased output of steam. If the adjustment were unalterable, the float, maintaining the same level, would leave a decreased quantity of water in the element. The adjustment is therefore useful, and, allowing a considerable range in the rate of output, is almost a necessary fitting.

Suppose the float to be falling with the water-level in the boiler, then, by means of the interior lever *EFH*, the rod *BH* rises and produces a fall in the rod *DK*, thus opening the valve *K*. This admits the feed water from the pipe *N* (from the feed pump) into the pipe *O*, which supplies the economiser. The water admitted to the economiser expels about the same amount from it into the feed pipe, and, through the generator feed valve, to the steam chest *V*, the pressure in the economiser being greater than in the generator.

A pressure considerably in excess of the working pressure (about double) is maintained in the pipe *N* by the feed pumps. The supply through the automatic feed regulator can be regulated by hand by a valve, *L*, which is usually fitted near the sediment chamber, but has no internal connection with it.

The lever *EFH* is pivoted on knife-edges at *F*, and is fitted with a roller bearing at *H*. All the packing glands of the lever rods are of special construction, and are generally packed with Belleville graphite packing. On the perfection of the packing and knife-edge bolts the efficiency of the apparatus is most dependent. The valve *K* can be examined by removing the screwed cap at the bottom of the valve box. At *X* a rod is fitted, so that the valve and lever can be worked by hand from the stokehold platform.

Water-Level.—A water gauge and test cocks are generally fitted on the float chamber, which acts as a steady pipe for this purpose. Another gauge glass is fitted to a steady pipe at the other side of the boiler, the upper and lower parts of which are connected in a similar way to that in which the float box is connected with the opposite wing element. The water-level should be only 1 to 2 inches in the gauge glass when lighting up, and if steam is required quickly, the level should be only just showing in the gauge. About one-half to two-thirds gauge is kept when working.

The water-level shown by the gauge does not necessarily show the amount of water in any element other than the one with which

it is connected. This applies to any element boiler with which the gauge is connected in the same way. A slight obstruction at the nipple connection on the feed collector (or any part of the channel) may prevent the circulation of water in the element to which it is connected, and lead to overheating.

The quantity of water which the boiler contains at different rates of combustion varies, although the same level is apparent in the gauge. This is the same in all boilers, and does not constitute a defect if the quantity of water and rate of circulation are sufficient to prevent overheating.

It has been found possible to increase the area through the nipple connections as experience has been gained in practical working. When the boiler is clean and in proper working order, the circulation has been found sufficient to prevent any suspicion of overheating for moderate rates of combustion, up to about 30 lb. of coal per square foot of grate.

If the flap valves in the sediment box fail to act, and do not thus prevent a reversal of current, it is possible, by inclining the ship more than the small inclination of the tubes, $2\frac{1}{4}^\circ$ to $2\frac{3}{4}^\circ$, to produce such a reversal of circulation. In naval vessels, therefore, the boilers are now so arranged that the tubes lie in the same direction as the keel of the vessel. This also applies to the Niclausse, Dürr, Babcock, and other boilers in a varying degree, according to the arrangements fitted and the inclination of the tubes to the horizontal.

Lime Strainer.—At the top of the hot water collector a strainer is fitted on the outlet pipe to prevent the passage into the generator feed pipe of particles of dirt or grease collected by the lime. Lime is mixed with the feed water to prevent any acidity of the water attacking the internal surfaces of the boiler, and as a nucleus for collecting particles of grease. A tank is fitted in the engine room, above the main feed tank, into which a certain proportion of lime is thrown and mixed with a small amount of the feed water taken from the main feed supply. The clean lime water is allowed to gravitate to the feed tank, whence it is pumped into the boilers with the feed water. The quantity of lime used should give the water sufficient alkalinity to turn the natural colour of litmus to *blue*; generally 2 lb. per ton of "make up" distilled water feed.

Soda should not be used in any boiler under ordinary conditions

of working. In addition to contributing to the formation of acid (hydrochloric acid particularly), it makes a priming mixture which is very difficult to cope with. Soda is, however, sometimes used for the purposes of cleaning the interior of the generating tubes and other internal parts. After its use the boiler should be carefully cleaned out with fresh water; or, if a sufficient supply of fresh water is not available, sea-water can be used for the earlier and rougher part of the operation of washing out.

Corrosion of Tubes.—If the system of supplying lime is efficient, and there is no doubt that it can be made so, the principal agent of corrosion is the oxygen always present in the air admitted with the feed water. Distilled water contains very little air in suspension (or, as is more probable, in solution), while fresh water, although much freer from air than sea-water, frequently contains a comparatively large amount. The heating of the water in the economiser sets free this air, and, as the oxygen contained in it is very probably in a nascent state, the oxygen readily combines with any metal with which it comes in contact. The early economisers suffered from corrosion particularly, and the economiser discharge pipe, or connection with the generator feed pipe, was frequently corroded away in a short time. A large proportion of the air is set free in the economiser, especially when higher rates of combustion are used and the economiser discharges water at a temperature of 200-220 degrees Fahrenheit. At ordinary rates of combustion, burning 15 to 20 lb. of coal per square foot of grate, the water discharged from the economiser has a temperature of about 150 to 170 degrees, and more air is carried into the generator.

As a protection from corrosion, a zinc angle, about 4 feet long, $\frac{3}{16}$ of an inch thick, and $1\frac{1}{4}$ by $1\frac{1}{4}$ inches wide, is placed with the angle uppermost in every economiser tube. In at least one tube of each generator element immediately above the apparent working water level, a similar zinc angle is enclosed in a galvanised wire netting case, so that particles of zinc may not be carried away by the movement of the water and steam and possibly choke free circulation. This protection, together with the abandonment of the use of soda, appears to have checked active corrosion.

Reducing Valve.—To reduce priming and the chances of its occurring, it was thought advisable to fit a reducing valve in conjunction with Belleville boilers. The general practice in some

foreign navies is still to fit reducing valves in conjunction with all types of water-tube boilers. In the British Navy the practice has been abandoned, because the supposed extra efficiency obtained by their use did not appear to justify the increase of weight and space involved. It still exists in some vessels, however, and a few engineers prefer their retention. The valve is fitted next the separator in each engine room. In some of the earlier ships a reducing valve was fitted to each boiler, which increased the number of fittings.

The effect of reducing the pressure between the boiler and engine is to produce a better quality of steam, as by lowering the pressure (and temperature) a certain quantity of heat is set free which converts into steam some of the particles of water in suspension. It also partly counteracts any condensation which may take place through imperfectly lagged steam pipes. If the steam generated by the boiler were *dry saturated* steam, the effect of reducing the pressure might very slightly superheat the steam. Within moderate limits this is not a probable effect, as nearly all common steam contains some priming water or particles of water in suspension.

Compared with other boilers the Belleville has a very small steam space, and the fitting of reducing valves at the engine ends of the steam pipes adds the space contained within the latter to the actual steam space of the boilers. The larger space, allowing a greater fluctuation of speed without producing priming, which is frequently due to lack of steam-room, acts as an accumulator of energy, and provides for a temporary increase, or decrease, in speed until the fires are adjusted to suit the new conditions.

A sectional elevation is shown of a reducing valve in Fig. 71. There is a cylindrical gridiron valve, *A*, which has ports cut in it, extending about two-thirds of the circumference. This valve is open at the bottom end and partially closed at the top, a few holes only being drilled to allow the reduced pressure access to the space above the valve. These holes are about $\frac{5}{8}$ inch in diameter. The valve *A* is connected by two loose pin joints with a spindle, *B*, which is connected with the lever *CDE* at *C*; this lever has its fulcrum at *E*. Similar ports are cut in a cylindrical valve face, *G*, which surrounds the valve, and which is secured into the valve chest *V* in a similar manner to that of the false face of a cylindrical slide valve into the slide casing. The lever *CDE* has a crossbeam

at *D*, poised on knife-edges, to which are attached frames containing several springs. The lower part of the frame is attached by a similar crossbeam, *H*, to a square-threaded adjusting screw, *L*, which passes through the boss of a wheel, *M*, working in a bracket on the bottom cover *N* of the valve chest. By means of the wheel *M* the tension of the springs can be adjusted. The end *E* of the lever *CDE* is connected with a support on the valve chest by a rod, *EF*.

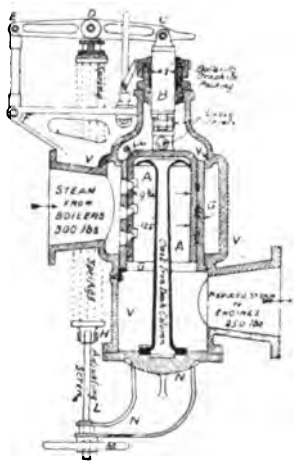


FIG. 71.—Belleville Reducing Valve.

In the figure the valve is shown *open*, the ports of the valve being opposite to those in the valve face. Steam enters from the steam pipe as indicated, and finds its way through the ports into the interior of the valve, impinging on the cast-iron *dash column* as it does so.

The valve is in equilibrium when the tension on the springs is balanced by the upward pressure on the valve. It will be noticed that there is an unbalanced upward pressure acting on the area of the spindle *B*, the upper end of which is open to the atmosphere. From this it will be seen that if the pressure exceeds that for which the springs are adjusted, the valve will rise, and consequently tend to close itself. By adding more springs to the frame, or by using the adjusting screw to put on more tension in the springs, the reduced pressure would be increased; and *vice versa*.

By careful adjustment, when the flow of steam is fairly constant through the valve, there is very slight movement or oscillation. As there are no packing rings round the valve, there is a small leakage at all times, and particularly when the engines are stopped; consequently, it is convenient to have a small safety valve on the *reduced* steam pipe to take away any excess of pressure over that required. A drain pipe connected with an automatic steam drain trap is also generally fitted to take away any water from the lower part of the valve casing.

Fusible Plugs.—The original Belleville boiler (without economiser) is fitted with a fusible plug in two junction boxes of each generator element, and this practice has been continued with

the economiser type, although possibly unnecessary with the improved circulation and larger openings at the nipple connections with the feed collector. In H.M.S. *Venerable* the Admiralty have sanctioned the permanent plugging of the lower row of plugs, and in H.M.S. *Drake* of the upper row. Fusible plugs have always been a source of trouble when high rates of combustion are used, and many devices have been suggested, and in some cases fitted, to prevent the plugs being blown out or melting. If pure lead, which has a melting-point of 600 to 625 degrees F. be used, and the plugs be mechanically compressed so as to produce a certain elementary stiffness, melting is not a very usual occurrence; but if tin be mixed with the lead it lowers the melting-point (although the plugs are stiffened and more easily rammed in place), and melting is probable under assisted draught. To prevent blowing out, the most satisfactory device appears to be to cut a slight groove at the mid-length of the plug hole, which, being only about $\frac{3}{16}$-inch in diameter, makes it therefore rather a difficult matter. The plugs require renewal at least every six months, and the usual practice is to renew them every time the boiler is opened for cleaning and examination.

In the author's opinion, fusible plugs are unnecessary in the elements connected directly with the water gauges; and also plugs in two or three of the elements (preferably just above the working water-level for convenience of renewal under steam) of each generator are sufficient for all practical purposes.

DÜRR BOILER.

Typical Features.—The generating tubes are straight, and inclined at an angle of about $10\frac{1}{2}°$ to the horizontal, and all are attached at one end to a single vertical chamber or header. The other end of each tube is free to expand through a hole in the casing, which supports the weight of the tube, but does not hold it. Each generator tube has another and smaller tube, as shown in Fig. 72, called the *circulating tube*, inside it and concentric with it, and the boiler is consequently of the *double concentric-tube* type. Each tube bundle forms an element, and can be renewed in a few minutes when the boiler is not under steam.

For British naval purposes one size of generator tube has been

used, $3\frac{1}{2}$ inches in diameter, and the inner or circulating tubes are $1\frac{3}{4}$ inches in diameter and $\frac{1}{50}$-inch thick (19 L.S.G.).

Above the boiler and attached to the steam collector are two rows of *dryer tubes*, of similar construction to the generator elements, but which only contain the upcast steam and a moderate proportion of water carried up with it. These dryer tubes form the internal passage-way of the steam from the collector into the steam pipe.

Path of the Furnace Gases.—The gases rise from the fire between the tubes, which are in vertical staggered rows. Baffles are fitted between the tubes horizontally and block the passage-way over about two-thirds of their length. These baffles are variously arranged, but in all cases the top one extends from the front end

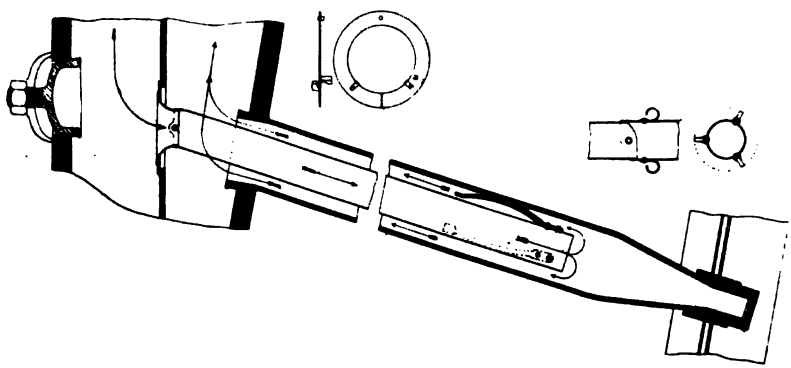


FIG. 72.—Dürr Tube Bundle.

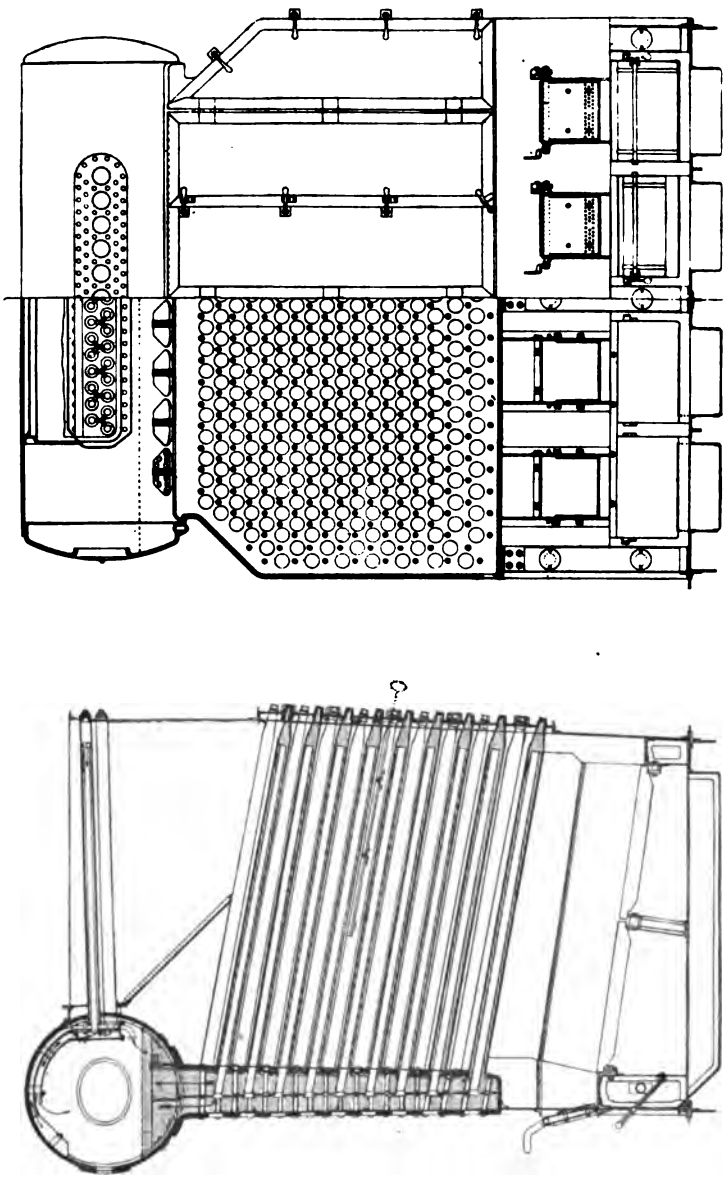
and covers about two-thirds of the passage-way from the generator. Fig. 73 shows a cross-section of the boiler, and Fig. 74 a front elevation in part section.

Brickwork and Baffles.—The brickwork is of the usual character, nearly surrounding the furnace, and preventing the direct impingement of the flames on the back end of the tubes and the bottom of the vertical collector.

The baffles are generally made of cast-iron plates, which can be moved by external rods or wires along the tubes on which they rest, and thus allow some variation of the passage opening. Some of these rods or wires pass through holes drilled through the horizontal stays fitted between the front and back plates of the vertical collector.

Vertical Collector.—The front ends of all the tubes are con-

nected with a single collector or header, built of steel plates front



FIGS. 73 and 74.—Dürr Boiler.

and back, with closed ends and bottom. The upper part is flanged

outwards, and is riveted to the steam collector, leaving a large opening between the two front and back plates, for the downward circulation and upward passage of steam into the steam collector. The front plate of the vertical collector is nearly upright, but the back plate slopes towards the stokehold platform at the bottom, thus leaving a larger opening between the two plates at the top end. For a large boiler the opening at the top is nearly 12 inches wide, and at the bottom nearly 8 inches. The circulating plate, to which the ends of all the circulating tubes are attached, is parallel to the front plate, and about 4 inches from it, thus leaving a larger space for the escape of the steam and hot water into the steam collector.

The front and back plates of the collector are stayed together by screwed stays, riveted over on the ends to make a pressure-tight joint. A stay is fitted in the space between each three tubes, and on each one a pair of nuts holds the circulating plate in position. The total pressure between the front and back plates of the collector is distributed over the stays.

The circulating plate is made in several pieces, forming one fair surface, so that it can be removed; but the butting edges are not entirely steam and water tight.

Tube Bundle or Element. The tube bundle, shown in Fig. 72, consists of three principal parts the outer or generator tube, the inner or circulating tube, and the back end closure.

The generator tube is enlarged at the part where it fits into the back plate of the collector, and coned to make a surface metal-to-metal joint with the coned seating in the tube plate. The coned seating is normal to the plate and, to produce the necessary inclination, the tube is bent just in rear of the cone. All the cones and seatings are standardised. The front end of the tube projects about half an inch into the collector. The tube is forced home on its coned seating by hydraulic pressure. The end pressure, due to the pressure in the boiler and the unbalanced area of the tube, retains the tube in place and tends to keep it pressure-tight.

The back end of the generator tube is drawn down in the form of a hollow cone, and a screwed bronze cap nut closes the opening. These nuts can be removed for cleaning and draining the tubes, and their removal does not at present appear to be attended with subsequent leakage. The face of the nut is made with two concentric V-shaped projections, which tend to produce a tight joint on the face of the collar fixed on the tube end.

The inner or circulating tube is made of thin sheet metal, about $\frac{1}{50}$-inch thick, and extends from about 4 inches from the back end of the generating tube nearly to the circulating plate. A bell-mouth is riveted to, and supports the end of the circulating tube, and is itself supported by a washer connected with the circulating plate. The washer is secured to the plate by an ingenious arrangement of four clips and socket holes, and the inner tube can be easily removed by lifting it away from the plate. The washer and flange of the bell-mouth block the passage-way between the downcast and upcast currents.

The back end of the inner tube is supported, clear of the generator tube, by three coach springs, each of which is riveted at one end to the inner tube.

The holes opposite the ends of the tubes, and through which they are passed into their places, are closed by inner closures with coned joints and seatings. The closures are hollow bodies, tending to expand in the holes under pressure, and, to prevent their expulsion from the boiler, are made with a slight flange. They are drawn home on the seatings by a strongback or dog; but as the pressure inside the boiler always tends to keep them pressed home, very little pressure is necessarily exerted by the dogs. The closures are standardised, but as they must be inserted from the inside of the boiler, are not quite so readily replaced as in the Niclausse boiler. The holes in a few vertical rows are made slightly larger, so that the closures for the remaining holes can be passed through them. The closures for the larger holes must be passed in from the steam chest.

Steam Collector.—The vertical collector is riveted to the steam collector, and forms one continuous whole with it. The part of the steam collector which is cut away to form the opening to the vertical collector is reinforced at necessary intervals by vertical steel plates riveted to the lower part of the shell. These plates are actually *tie* rods across the opening from front to back, and resist the tendency to straighten the circular shell of the steam collector when under pressure.

The *dryer tubes* are fitted into coned holes at the back of the collector, slightly above the horizontal diameter, and the shell plating is strengthened by a doubling plate at this part. The dryer tubes are nearly horizontal, and are fitted in two staggered rows. The securing arrangements for both the inner and outer tubes are

similar to the other tubes in the boiler. A thin sheet baffle is fitted, as shown, through openings in which the rising steam, and any water carried with it, pass into the circulating or inner tubes. From the inner tubes the steam passes away through the annular space between the two tubes, thence to a practically steam-tight space, formed between the circular shell of the steam collector and another baffle, which is in connection with the steam pipes and safety valves.

The dryer tubes are fitted at a slightly varied angle for each row, so that only one row of holes is necessary in the steam collector shell, opposite to them, for their withdrawal. These holes are closed by internal hollow coned doors of similar construction to the doors fitted on the vertical collector. A doubling plate is fitted to the shell in wake of the holes to maintain the local strength.

The vertical circulating plate is continued upwards into the steam collector, and its top edge is just above the water-level.

Circulation.—At working height the water-level is a little above the bottom of the steam collector, and the whole of the vertical collector and generator tubes are therefore submerged, leaving the top part of the steam collector and the dryer tubes above the working level.

The feed water is pumped into the lower part of the steam collector, and flows over a baffle into the downtake part of the vertical collector. The baffle is fitted for the purpose of collecting any sediment and for warming the feed water. The supply of feed water is generally automatically regulated. The water descends into the downtake, passes into the inner tubes, and emerges from them into the back end of the generator tubes. The heating of the latter promotes a circulation towards their upper ends. The mixed steam and hot water are delivered into the uptake part of the vertical collector, and thence into the steam collector. The steam rises into the upper part of the collector, while the water flows over the top of the circulating plate into the downtake, and, mixing with the other water, commences another circuit of circulation. The steam, with a slight mixture of particles of water carried up with it, passes through slits or holes in the upper part of the baffle which thus forms an internal steam pipe, into the inner dryer tubes, thence through the outer tubes and baffle space into the steam pipe or safety valve.

NICLAUSSE BOILER

Typical Features.—The generating tubes are straight, and inclined at an angle of about *six* degrees to the horizontal, and are

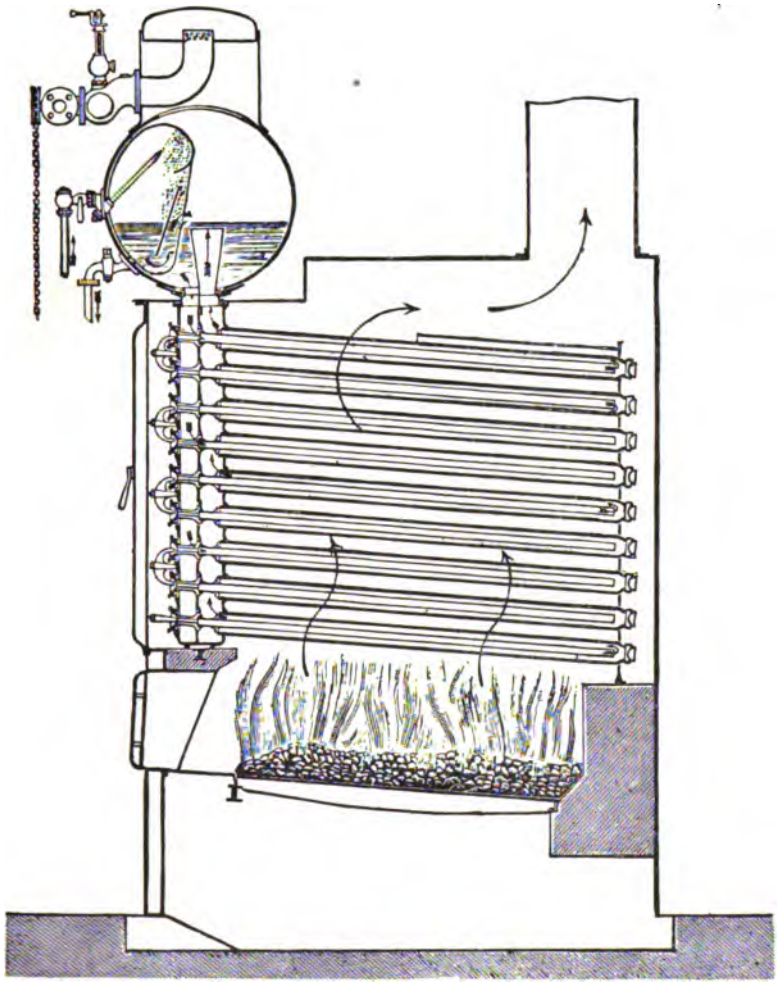


FIG. 75.—Niclausse Boiler.

attached to vertical headers at one end only, as shown in Fig. 75. The other end of each tube is free to expand through a hole in the casing, which supports the weight of the tube, but does not hold it.

Each generator tube contains another and smaller tube, called the *circulating tube*, inside it and concentric with it. This arrangement is common to the Dürr and other less-known boilers, and is called the *double concentric-tube* type. Each tube bundle forms an element, but a number of these elements joining into one vertical header is also termed an element. There are in each boiler from 10 to 20 such elements.

For British naval purposes only one size of generator has been used—3·3 inches in diameter, with an inner circulating tube of $1\frac{9}{16}$ inches diameter. For foreign navies, tubes of smaller diameter are used.

Path of the Furnace Gases.—The gases rise from the fire between the tubes, which in British warships are in vertical staggered rows; but for some foreign types the tubes are very little staggered, and attached to vertical headers with straight sides, as shown in Fig. 77. Baffles are fitted in some cases horizontally between the tubes, and in all cases a single baffle is fitted above the top row of tubes to check the otherwise short cut to the funnel, and reduces the passage-way at this part.

A gas-mixing apparatus, similar to the Belleville arrangement, is fitted to some recently constructed boilers of this type.

Brickwork.—The brickwork is of the usual character, but special means are taken to prevent the direct impingement of the flames on the back ends of the tubes, and also to protect the bottom of the vertical headers from similar action.

Sweeping Tubes.—Tubes can be swept without any difficulty from the stokehold platform by inserting wire brushes and scrapers between the vertical collectors. The freedom of the tubes from any external obstruction also conduces to easy cleaning. Smoke-box doors are fitted, both at back and front of the boiler, and great care should be taken to keep them perfectly air-tight under steam. For this purpose a strand of asbestos is generally used to secure a tight jointing of the doors at the back end, and in front the spaces between the vertical headers are generally packed with asbestos rope.

Circulation.—At working height the water-level is below the horizontal diameter of the steam collector; the whole of the boiler is therefore submerged below this point, leaving above the working level only the upper part of the steam collector and a small steam dome, rising from the collector.

The feed water is pumped into the steam collector, and is generally automatically controlled by the Niclausse feed regulator. The water descends into the front part of each vertical header, similarly to the Dürr boiler, which is screened off from the back part by a vertical circulating plate. This plate holds the end of the circulating tubes, leaving them open to the front part of the vertical header, which forms the downtake for all the circulating tubes. The water then passes into the circulating tubes and along them until it reaches nearly the end of the generating tube. The circulating tube is rather shorter than the generating tube. The heat from the fire and furnace gases passes into the water from the outer tube, and the increased temperature and generation of steam causes a circulation in the direction of the upper end. The generating tube delivers its contents into the space between the circulating plate and the back face, or tube plate, of the vertical header. Thence the steam and water rise into the steam collector, and are delivered through a bell-mouthed vessel just below the working level. The steam rises into the upper part, and the water takes up another circuit of circulation.

Steam Collector and Dome.—The steam collector, shown in Fig. 76, is a hollow cylinder, stretching across and just above the vertical headers, which are next the stoking platform. The steam dome tends to produce dryer steam, which in rising to a higher level sheds some particles of water on its passage. The dome is of small size, and some of the boiler mountings, such as the steam valve, safety valves, and pressure gauges, are connected with it. A long stay is sometimes fitted from end to end of the steam collector.

The bottom part of the collector is strengthened by a doubling plate to receive the upper cone of the double-coned joint, which is made with each vertical header. A coned hole is made in this thicker plate and also in the top end of each vertical header. A short length of tube, coned at each end, fits into these coned seatings, and the double joint is forced home by four coned-headed bolts, passed in from the steam collector and nutted below a flange formed on the top of the vertical header. On the top of this cone a sleeve, divided into two parts by a plate of suitable shape, is fitted, and rests on a flange. The upper part of this sleeve is bell-mouthed above the uptake of the collector, and the top end of the opening is just below the working water-level. A projection on

the dividing plate, below the sleeve, connects with the circulating plate, and makes the division continuous. A lug projects from the sleeve and fits over one of the bolt heads, which is left long enough for the purpose, and secures the sleeve piece and its several connected parts in their place inside the boiler.

The feed water is injected upwards, in the type shown, and is afterwards deflected downwards by a baffle or separating plate into a settling chamber, which is formed by a trough with its upper

DETAILS OF NICLAUSSE BOILER.

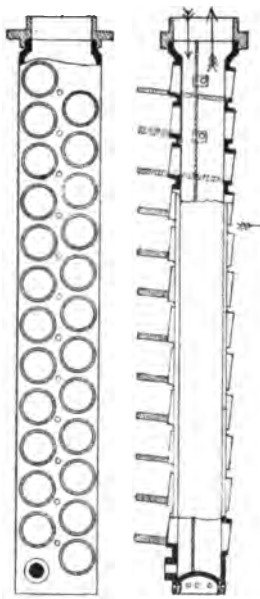


FIG. 77.—Niclausse Straight-sided Header.

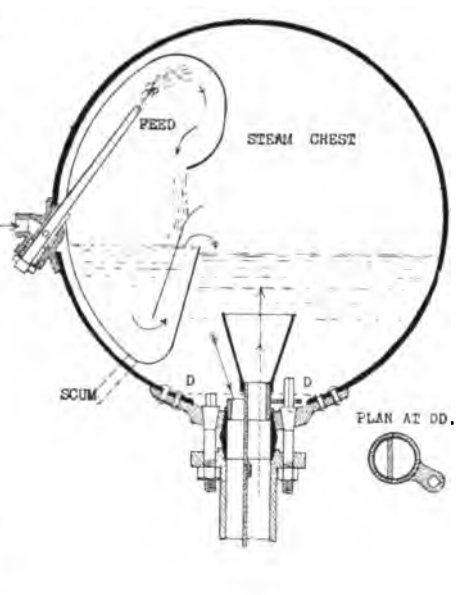


FIG. 76.—Niclausse Steam Drum.

edge just above the working water-level. The water spray is warmed on its passage, and some of the sediment contained in it is deposited in the settling trough; from the settling chamber the sediment is blown out through the scum, or surface blow-down valve. The hot water from the trough overflows into the downtake.

Vertical Collector.—There are two types of vertical headers, one with straight and the other with sinuous sides. The straight-sided one, shown in Fig. 77, was introduced by MM. Niclausse, so that forged steel could be used for it in place of cast-iron; but as

it has since been found possible to manufacture the sinuous type of forged steel also, the straight-sided header is not now fitted. With the straight-sided header the tubes in contiguous elements are in straight vertical rows, leaving a straight passage for the furnace gases, and the whole of the tube surface is not brought into play. In the sinuous arrangement (Fig. 78) the elements are zigzagged into each other, and all the tubes present a staggered passage-way for the furnace gases.

FIG. 78.—Niclausse Element.

The header is divided into two parts—front and back, downtake and uptake—by a thin vertical circulating plate. Greater space is allowed between the back and circulating plates for the hot water and steam passing through it than for the cooler water passing downwards between the front and circulating plates to the inner tubes.

The upper end of each header is connected with the steam collector by the double-coned joint already described, but the lower end is free and closed. A small pipe, about $\frac{3}{8}$-inch internal diameter, is connected with the downtake space at the bottom of each header, and leads into a sediment pipe running along the front of the boiler. This sediment pipe or mud-drum is connected with the blow-down arrangement.

With the sinuous arrangement of vertical header, stays are not usually required; but the straight-sided one is stayed from front to back by screwed stays, which also serve for studs, on which the dogs, securing the tubes in place, are fitted. The circulating plate is secured in position by small angle brackets, and by the connection with the delivery plate above it.

Tube Bundle or Element.—The tube bundle consists of three principal parts—the outer or generator tube, the inner or circulating tube, and the end closures.

The generator tube, shown in Fig. 79, is enlarged at the part where it fits into the back plate of the vertical header, and coned to a taper of 1 in 25, to make a surface joint with the hole in the plate. This enlargement continues through the hole in the circulating plate, which is slightly larger in diameter than the tube, and it is therefore not a perfect fit. From this point to the front end of the tube it is reduced to a smaller diameter, and an external screw thread, 12 threads to the inch, is cut on it. Over this end a

specially shaped ring is fitted, making a pressure-tight joint with the screw thread and the butting end of the tube. The outer part of the ring is coned to 1 in 40, and makes a surface joint with the front plate of the vertical collector, or front tube plate. The ring B is hollow between the screwed and the coned joints, and as this space is open to the pressure contained in the boiler, the tendency is to produce an outward pressure on the coned joint and a corresponding pressure-tightness. The external part of this ring is made octagonal (8 sides), for use with a spanner, and the inner part is fitted with a screw thread, 12 threads to the inch.

The inner or circulating tube is made of thin sheet steel, about

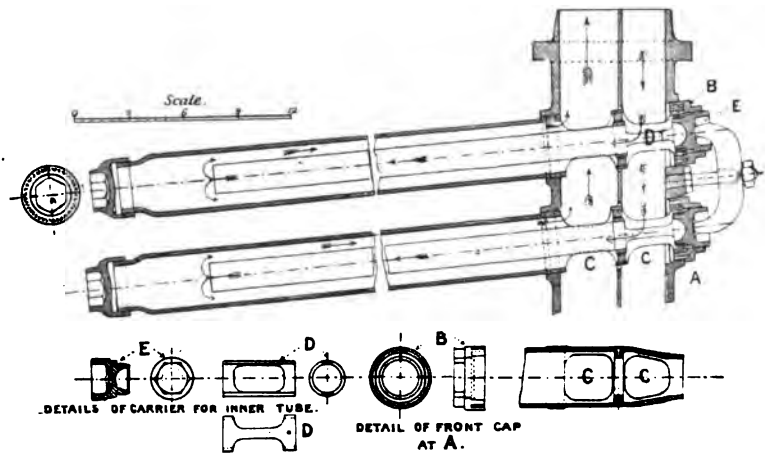


FIG. 79.—Niclausse Tube Bundle.

$\frac{1}{50}$-inch thick, and extends from about 4 inches from the back end of the generator tube to the circulating plate. About this part it is bell-mouthed, and riveted to a thicker tube of larger diameter. The annular space between the two tubes, the generator and circulator, is blocked by a fillet ring riveted to the outer tube. These arrangements block the free passage-way between the two sides of the circulating plate. The front end of this extension of the inner tube is screwed with an internal thread, and the front-end tube closure is screwed into it, 16 threads to the inch.

The front-end closure is cupped towards the boiler, and receives, on its outside, the inner tube, as already stated, and makes a screwed joint with the end ring. Part of the hollow is made hexagonal (six sides), and is used for the reception of the spanner

when screwing up the joint, and another spanner is used at the same time to prevent the ring being disturbed on the coned seating. This is useful when removing the inner tube, as it leaves the coned joints undisturbed on the front and back tube plates.

Both the coned joints are subjected to internal pressure, which assists in keeping them pressure-tight. Any slight error in the correct distance between the two internal and external cones can be adjusted, in the type described, by altering the position of the butting end of the tube and ring B.

Vertical passage-ways C, or *lanternes*, are cut in the generator tube, between the tube plates and circulating plate, to allow for entry and exit of the steam and water. The extension length D of the inner tube is also cut away to provide a vertical passage-way for the downtake water into the circulating tubes.

At the back end the inner tube is supported by a cast-iron ring with three or four radial arms, the outer edges of which are in contact with the inner surface of the generator tube. This arrangement allows a free passage-way for the circulating water into the space between the two tubes.

The back end of the generator tube is reduced in diameter, a screw thread is cut on it, and a cap nut fitted outside. This cap nut is of slightly less external diameter than the body of the tube, so that it need not be removed when withdrawing the tube from the boiler. A square or hexagonal socket, or a series of slits, is fitted outside the cap, so that a spanner can be used for screwing up or removing it. The exterior casing supporting the tubes at the back end is made in separate strips for each vertical header element, to facilitate removal.

A tube bundle, or element, can be renewed in a few minutes, as all the cones are standardised for both size and distance. There is some difficulty in draining the boiler entirely, unless a pump or syphon is used for each tube in turn. If sufficient space is available, the back ends of the tubes might be removed; but the process is very long, and the difficulty of ensuring tight joints without refilling and testing the boiler is evident.

Although the inclination of the tubes is small, great rates of forcing have been safely used with this type of boiler *on shore*. For marine purposes the tubes must be placed in line with the keel, and the angle of inclination could be increased with advantage to both circulation and safety.

Fig. 80 shows a tube bundle fitted in some of the earlier types of Niclausse boilers.

To hold the tube firmly against the coned seatings, a screwed stud is fitted between each two tube ends, on which a strongback, or dog, is fitted, with its ends butting against the ends of the tubes or closures.

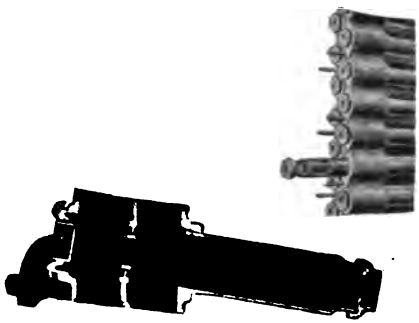


FIG. 80.—Niclausse Tube Bundle (old type).

As the tube is only rigidly connected at one end, and the other end is free to expand lengthways, the allowance for expansion is perfect. The variation in distance, due to any difference of temperature between the two cones on the tube and their seatings in the tube plates of the vertical header, is probably very small, as both parts are in free contact with the same steam and water. In practice, a good joint under steam is obtained.

Axial Tension in Water-Tubes.—In all hollow bodies subjected to internal fluid pressure there is a pressure equally distributed on each unit of area of interior surface, and this pressure is exerted in an endwise direction as well as radially. In the Niclausse and Dürr boilers, where one end of each generating tube is closed, there is an end-pressure or thrust tending to force the tube away from its attachment to the header or other part of the boiler. The particular form or shape of the body to which the tube is attached does not alter the principle, and the end thrust always = pressure × area on which such pressure is exerted. This area always = smallest area of the hole through which the tube attachment is made, which may generally be taken as that corresponding with the exterior diameter of the tube.

In the *Niclausse* boiler the internal pressure is exerted on an area corresponding to the hole through the back tube-plate; this pressure tends to hold the tube in place. The front closure fitted opposite the end of each tube, being in rigid connection with the tube, is subjected to an internal pressure which acts in the opposite direction and tends to eject the tube. The two forces are nearly balanced, but the end thrust is slightly in favour of

ejection, which is prevented by the dog-stay fitted between each pair of tubes.

In the *Dürr* boiler there is no rigid connection between the front closure and the tube, and consequently the end-pressure is principally exerted in the direction required to keep the tube in place on its coned seating.

In the *Belleville* boiler and economiser the junction boxes opposite to the area of each tube, and opposite to each other, support a mutually balanced pressure in opposite direction and approximately in line with the axis of the tube. The total end-pressure or thrust is taken on the screw thread securing the tube ends and ferrules. It is almost a practical impossibility to produce an absolutely perfect fitting thread commercially, and it may be considered that the apex of each thread will leave a slight cavity in the recess in which it rests. Any such tendency to leakage is increased by the end-pressure or load, and under working conditions a very slight percolation—it can hardly be dignified by calling it leakage, although in the aggregate the leakage is sometimes very serious—takes place along the spiral cavity of the screw thread. If the graphite packing and nut are closed into a good tight joint, the percolation is stopped when the end-pressure is not supported at the joint. At the junction of the sleeve and ferrule the compensating effect of the nut is lost, and the end thrust, widening the space between the sleeve and the junction box, tends to allow leakage through the spiral cavity.

In the *Babcock and Wilcox* boiler and in all other water-tube boilers in which the tubes are rolled and expanded into headers or other corresponding parts, the end-pressure is exerted on an area equal to the smallest area of the tube-plate hole and tends to withdraw the tube from its attachment. Such possible withdrawal is now generally prevented by making a bell-mouth on the projecting tube end: as shown in Fig. 54 for the Yarrow boiler, and in Fig. 51 for the Laird boiler. Particulars of a withdrawal test are given at page 109.

In all the water-tube boilers described in this book, with the exception of the Niclausse and Dürr types, every tube for all practical purposes must be considered as a stay-tube, and precautions taken accordingly.

Assuming a working pressure of 300 lb. per square inch, the withdrawal force, or axial tension in the tube, is about the following:—

Belleville, generator tubes,	5″	diameter,	5892 lb.		
„ „ „	$4\frac{1}{2}$″	„	4770 „		
„ „ „	4″	„	3770 „		
„ economiser tubes	$2\frac{3}{4}$″	„	3564 lb.	}	pressure, about 600 lb.
„ „ „	2″	„	1884 „	}	per square inch.
Babcock, boiler generally	4″	„	3770 „		
„ small tubes	$1\frac{13}{16}$″	„	800 „		
Yarrow boiler	$1\frac{3}{4}$″	„	723 „		

and others in proportion.

Size and Arrangement of Modern Boilers.—In the largest modern warships for the British Navy there has been a distinct tendency to standardise the size and steam production of the boilers, which are of either the Babcock and Wilcox or the Yarrow or the White-Forster types. Each boiler is capable of producing about 20,000 lb. weight of steam per hour when worked at full power, which for the Babcock type means burning about 30 lb., and for the other two burning about 40 lb. per 1 square foot of grate per hour; but the total weight of fuel burnt is about the same for each boiler. The I.H.P. obtained depends almost entirely on the steam consumption of the engine per unit of power, and if auxiliary purposes be included it may be considered that each boiler is capable of producing steam for about 1000 I.H.P. or 1250 shaft H.P. for all purposes. (In torpedo boats, using oil fuel and turbines, about 2000 I.H.P. is obtained per boiler.)

The best arrangement is to place the boilers back to back; the furnaces fore and aft; to have a bulkhead between each stoking space so that the boilers facing each other shall be divided by a transverse bulkhead as shown in Fig. 228, and thus prevent the undue fatiguing of the stokers from a fire at their front and back; and to make, as far as possible, each boiler-room a self-contained water-tight compartment. The fore and aft length of such a boiler-room containing six boilers is about 38 feet, and its width for Babcock and Wilcox boilers about 45 feet; but with Yarrow or White-Forster boilers the width is about 3 feet less, allowing for a 30″ passage way between the two stokeholds of the same boiler-room.

The same number of boilers (18), of exactly similar construction and steam producing power, are fitted in H.M.S. *Agamemnon* and *Dreadnought*; but while the Navy List credits the former with only 16,500 I.H.P., the *Dreadnought* has obtained 22,000 shaft horse power by the economy of the Parsons turbines.

Influence of Practice on Construction and Design.—The grate area and the heating surface, on which the output of steam from any boiler is almost entirely dependent, are given for several types of the more modern boilers in the following table, which should be compared with actual trial results :—

Type of Boiler.	Maximum Coal Burnt per 1 sq. ft. of grate area.	Maximum I.H.P. per 1 sq. ft. of grate area.	Heating Surface per I.H.P.	Heating Surface. Great Surface.
	Lb.		Sq. Ft.	Ratio.
Small tube, for destroyers . .	50	20·0	2·25	45·0
Small tube, $1\frac{1}{8}$″ to $1\frac{1}{2}$″, small cruisers and scouts	50	20·0	2·6	52·0
Yarrow, White-Forster, $1\frac{3}{4}$″ tubes .	40	20·0	3·0	60·0
Babcock, $1\frac{1}{16}$″ and $3\frac{1}{16}$″ tubes (two rows)	30	12·9	2·7	34·8
Babcock, $3\frac{3}{16}$″ tubes . . .	30	12·9	2·6	33·8
Niclausse, 3·3″ tubes . . .	30	12·9	2·6	33·8
Dürr, 3·25″ and 3·5″ tubes . .	30	14·0	2·6	35·0
Belleville 4″, and Economiser 2″ tubes	30	13·0	2·6	32·5
Cylindrical, mercantile type . .	25	14·0	$2\frac{3}{4}$ to 3	39 to 42
,, H.M.S. *Majestic* . .	32	15·2	2·0	30·4
,, H.M.S. *King Edward VII.*	30	15·0	2·3	34·2
,, H.M.S. *Africa* . .	30	15·0	2·3	34·3

The heating surface actually fitted is generally slightly in excess of these specified quantities, but the excess is of little consequence. On the other hand, the rate of combustion, or quantity of coal burnt per square foot of grate area, is seldom reached under modern trial conditions, but was frequently exceeded previous to 1898, especially in destroyers and torpedo craft. The introduction of a limit to the coal consumption allowed per I.H.P. per hour on contractor's trials has had a most marked effect in reducing the normal coal consumptions of naval vessels, and steady improvement is still being made.

The installation of the following types in the British Navy ceased in favour of improved water-tube types :—

Belleville in 1900.

Cylindrical in 1905 (fitted in conjunction with various water-tube types).

Babcock, $3\frac{3}{16}$″, in 1902.

Niclausse in 1904.

Dürr in 1904 (fitted in three ships, and those of the *Medusa* are transferred to the *Vulcan*).

In the above table the maximum I.H.P. per 1 sq. ft. of grate is less for a mercantile cylindrical boiler than for a similar boiler fitted for naval purposes, and the maximum rate of combustion is also still further decreased because the steam consumption per I.H.P. is also less.

The *length of furnace* grate should not as a rule exceed either $6\frac{1}{2}$ feet or three times the mean height (or semi-diameter for cylindrical furnaces). In recent cylindrical naval boilers the actual length is 6′ 9″, and mean diameter about 49 inches (*Hindustan*); but in ships just previous to these the length is 5′ $9\frac{1}{2}$″, and diameter 38″ (*King Edward VII.*). The larger diameter is more efficient, and can be more easily stoked because the height of furnace is greater.

The *thickness of furnace* plating of the better corrugated types (Fox, Morison, Deighton-Morison, Brown cambered types) is given by the following formulæ, which are based on practical experience rather than any exact theoretical considerations. Owing principally to the improvement in manufacture, and the more nearly correct circular form so obtained, the maximum thickness allowable is now about $\frac{3}{4}$ inch. This thickness is fitted in the *Hindustan* mentioned above.

Board of Trade: Thickness in inches

$$= \frac{\text{Working pressure} \times \text{outside diameter at bottom of corrugations (inches)}}{14{,}000}.$$

Lloyd's Thickness in $\frac{1}{16}$ths inch

$$= \frac{\text{Working pressure} \times \text{greatest diameter in inches}}{1259} + 2.$$

The *steam space* may be less with high than with low pressure, because the total weight of steam contained in the boiler is greater per cubic foot of space. For naval cylindrical boilers the space is generally between 0·25 and 0·35 cubic feet per I.H.P. (full power understood), and the water-level is then at about one-fourth the diameter of the boiler from the top. In the mercantile marine as much as 0·4 to 0·5 cubic foot is obtained. As naval engines require about 3 or 4 lb. weight of steam more per I.H.P. per hour than those in the mercantile marine, the actual difference is still greater than appears from the above figures. For water-tube boilers the steam space is much less, partly because of the higher pressures generally used; but in all later vessels the space has been considerably increased and a greater efficiency has been obtained,

because the formation of wet steam is less liable to take place with the increased water surface area and steam space.

Having calculated the general requirements of grate area, heating surface, steam space, diameter of furnaces (which should be of the same length and diameter), and using some previous design for convenience of comparison, it is not difficult to obtain the general outline dimensions by a system of trial and error correction.

The calculation of the *thickness of shell plates* of a cylindrical boiler (or of a similarly shaped part of a water-tube boiler, such as the steam and water chambers) is based on the considerations given below :—

$$\left.\begin{matrix}\text{Test pressure} \times \text{internal diameter}\end{matrix}\right\} = \left\{\begin{matrix}\text{Thickness of metal} \\ \text{(in inches)}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Strength of metal} \\ \text{(in inch-tons)}\end{matrix}\right\} \times \left\{\begin{matrix}\text{Efficiency} \\ \text{of joint}\end{matrix}\right\}$$

The test pressure = $1\frac{1}{2}$ times the working pressure ;
Strength of material = Elastic limit, which for this case in tons per 1 square inch
= $\frac{1}{6} \times 28$, for cylindrical boilers (plates *not* exposed to flame) ;
= $\frac{1}{8} \times 24$, for water-tube boilers (plates exposed to flame) ;
Efficiency of joint = 75 to 86 per cent about, but is dependent on kind of joint used and riveting employed.

Parts of the boiler not of cylindrical section require special calculation ; and for the cylindrical type the stays have been already mentioned. For water-tube boilers, the feed collectors of the Belleville and Babcock types, the headers in the Babcock and Niclausse, the large water chamber of the Dürr with many stays, and the lower tube plates on the water chambers of the Yarrow and White-Forster boilers, are instances of methods used to obtain the necessary local strength.

The *maximum stress* allowable *on stays* at test pressure is dependent on the quality of the material (this is always the case for any stress), and in the most recent practice is limited to :—

18,000 lb. per 1 sq. in. of section for an effective diameter of *over* $1\frac{1}{2}$ ins. ;
16,000 lb. „ „ „ „ of $1\frac{1}{2}$ ins. or *less*.

The effective diameter is the smallest diameter at the bottom of the thread or at any smaller diameter of the stay.

The *thickness* of the *front and back plating* of a cylindrical boiler and its combustion-chamber plating (except the bottoms which are now made semicircular to avoid the otherwise necessary staying) are generally dependent on practical considerations for convenience of access. The stays are therefore pitched at certain distances from each other, and generally they, as well as the tubes, should be in vertical rows to allow easy cleaning with manual tools. The

longitudinal stays are usually pitched vertically above rows of tubes for this reason, but, as will be noticed by referring to Figs. 26 and 27, this is not always the case. The rules relating to the thickness and staying of flat surfaces are given in the Board of Trade or other rules, and are always complied with; being based on previous successful practice, it is usual to follow some previous design in this respect and then to check by one or both rules.

CHAPTER IX

BOILER FITTINGS AND MOUNTINGS

Furnace Fittings.—A general idea of the firegrate, which is the most important of the furnace fittings, can be gathered from the various figures of boilers in Chapters VI., VII., and VIII.

For a circular furnace, such as fitted in tank boilers, the fire bars are in two or three tiers, and extend, in naval practice, from just inside the fire door to near the beginning of the combustion chamber; but for the mercantile marine, the back end is shortened by 1 to $1\frac{1}{2}$ feet. Near the fire door the grate is somewhat above the horizontal diameter of the furnace, and is inclined downwards as it recedes, and at about its mid-length is very nearly central. The firegrate is inclined to obtain a better supply of air throughout, while by keeping it as low as possible a larger space remains above the fire for combustion. The air entering below the fire passes through the ashpit door, and is diffused into the fire from between the bars until at the back end only a moderate amount, commensurate with the contracted area, is admitted for combustion. The largest volume of the furnace gases passes over the fire at the back end, and the greatest area is allowed for it. Thus the area of opening for the supply of air and the space above the fire are the maximum amounts within the space allowed by the diameter of the furnace.

The *fire bars* rest on the bearer bars, which are placed across the furnace below the nearly butting ends of the tiers of fire bars. The ends of the bearer bars rest on lugs riveted to the sides of the furnace, and they can be lifted out after the fire bars are removed. The front and back ends of the fire bars rest on *dead plates*—so called because they give no life to the fire, no air being supplied through them. Sufficient side play for the fire bars, and

end play for both fire and bearer bars, must be allowed, to prevent any distortion of the furnace and its fittings by their expansion.

The back end is faced with *brickwork* to prevent the dead plate and its supports from burning away. Sufficient area—about one-fifth that of the firegrate—is allowed over the brickwork for the passage of the furnace gases through the throat of the furnace; an arch is sometimes used as shown in Figs. 113 and 114, to protect the riveted joint of the upper part with the combustion chamber. This arch is not necessary for ordinary rates of combustion; but when burning more than 25 lb. of coal, or 15 lb. of oil, per square foot of grate per hour, it is advisable to fit one.

The *furnace door* of cylindrical boilers is fitted in a wrought-iron or forged steel frame, nearly semicircular in shape, and flanged to cover the projecting end of the furnace, to which it is usually secured by a few bolts screwed through it radially to the furnace; the lower part, which forms the dead plate, rests on lugs fitted to the sides of the furnace. The door is made of dished plate, and fitted with an inside guard plate, secured at a distance of about 2 inches from it. The plate next the stokehold is perforated with a number of small holes, or slits, from $\frac{1}{2}$ to $\frac{3}{4}$ inch in diameter or width. The plate next the fire has an unperforated surface, and allows a sufficient margin for the air to pass, from the holes, round its edge into the space above the fire. A sliding door, or louvre, is fitted to shut off the air-supply through the holes when not required.

The door swings on a vertical hinge, and can be clipped either open or shut as required. Many fire doors are fitted on the self-closing swing-gate principle as shown in Figs. 48, 50, 44; that is, the lower pivot of the hinge is brought away from its vertical position below the top pivot, both sideways from the door and outwards from the boiler. There are many varieties, but the one described above is perhaps the simplest and best. The principal requirements are: (*a*) proper supply of air above the fire—about one-fourth might be supplied in this way; (*b*) the fire door should be capable of being locked open in a sea-way, and also be self-closing when required; (*c*) the interior plate or plates should be easily renewable, and should prevent the actual door from warping or burning away.

Fig. 81 shows a section through the door and frame of a Yarrow boiler, and a front elevation of the furnace door with its surroundings

removed. Air is admitted above and below the frame, and through an opening with a shutter in the middle of the door.

The *ashpit door* of cylindrical boilers is almost semicircular in shape, and entirely covers the opening below the dead plate. It is generally swung on horizontal hinges resting in lugs fitted at the front of the furnace, and when opened the lower part enters the furnace at the bottom. A rack is provided to keep the door in any required position; this arrangement is used for regulating the air-supply below the fire. The hinges are placed above the centre of gravity,

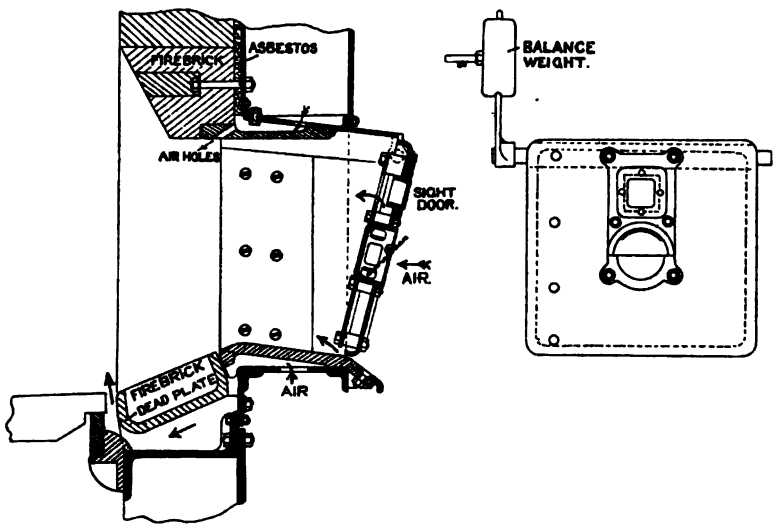


FIG. 81.—Furnace Door and Fittings.

so that the door closes naturally when not held open. Handles are fitted and the side lugs arranged so that the door can be lifted away, when required, for pricking the fire, raking out the ashes, etc. A removable bar is fitted across the mouth of the ashpit for use with the pricker, and is known as the *pricker bar*.

Automatically *self-closing ashpit doors* are generally fitted to boilers worked under very high rates of combustion, as in destroyers. The door is slung by three or four links of chain to a pair of gallows inside the casing; and the relative positions are such that when the pressure inside the ashpit is greater than in the stokehold, the door closes and is pressed against the inner face around the opening. Unless the forced draught is unnecessarily great, the ordinary door

opens an insufficient amount, and consequently a larger one is sometimes fitted inside the front of an airtight casing built out from the ashpit; this permits a larger opening, and thus increases the air-supply with the same air-pressure.

Ash-pans and Water Service.—The ash-pan is sometimes used for corrugated furnaces. It then consists of a thin plate, generally zinced by the hot process, bent to cover the lower part of the furnace below the fire; the edges are lipped, or fitted with angle steels which are lipped, to fit into the corrugations, and to prevent ashes and moisture from getting into the space between them and the furnace. The ash-pan facilitates the withdrawal of ashes, and its smooth surface presents less frictional resistance to the flow of air to the fire. Their continued efficiency is doubtful; they easily buckle from the heat of the fire, when the ashes and moisture have free access below them, and conduce to the corrosion of the furnace.

With water-tube boilers the ash-pan is necessary to prevent the ashes falling into the bilge. For high rates of forcing a little water is run into the bottom of the ash-pan, and prevents the overheating of the bars by immediately cooling the ashes, and thus conduces to continued efficiency. For corrugated furnaces, when used in this way, a lip is formed front and back to contain the water in the ash-pan.

The water is supplied by two pipes, one on each side of the furnace, which are perforated with a lot of small holes in such a way that the jets of water strike the lower edges of the fire bars below the whole surface of the grate. The pipes are provided with shut off cocks, and these are only opened when actually required for short intervals; a depth of 1 or 2 inches is generally sufficient for all ordinary purposes.

Position of Boiler Mountings.—In Figs. 82 and 83 the positions of the mountings usually fitted to a marine boiler are shown for cylindrical (tank) and Babcock and Wilcox (water-tube) boilers respectively. Similar mountings are seen in Figs. 48, 58, 55, 56, for upright water-tube boilers. The reference to the mountings shown in Figs. 82 and 83 is as below:—

1. Safety valves, at least two to each boiler; spring loaded with spindles vertical.
2. Steam valve, self-closing type, with spindle horizontal; connected with an internal steam pipe.
3. Pressure gauges, one graduated to rather more than the test pressure.

4. Water gauges; one glass tube type, and generally one plate glass—at least two.
5. Hydrometer cock, for testing density of water.
6. Auxiliary feed valve, on left.
7. Main feed valve, on right; if out of reach, gearing is fitted to control each feed valve (see Fig. 48).
8. Brine or scum valve, with internal scum pan just below the working water-level.
9. Manholes, and doors.
10. Air cock.

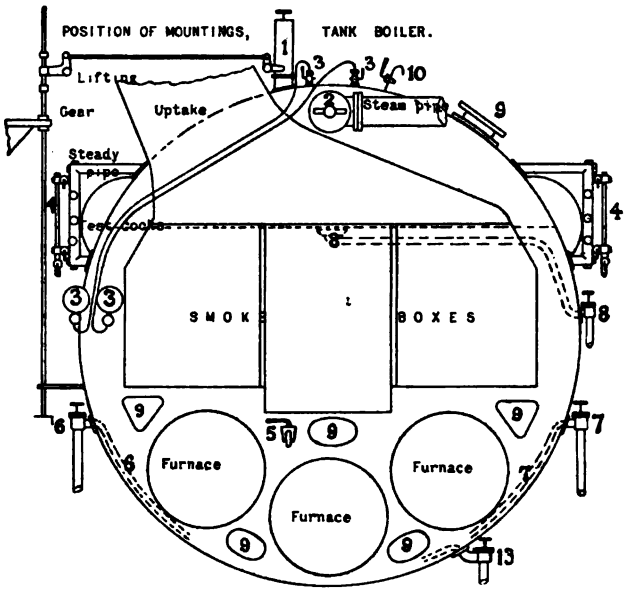


FIG. 82.—Mountings on Cylindrical Boiler.

11. Downcomer pipes.
12. Sediment chamber; a drain or running down valve is fitted to one sediment box of each boiler.
13, 14. Blow-down cocks.

The construction and other particulars are detailed under their various headings. All valve boxes and gauge mountings are made with spigots projecting into the boiler.

Water Gauges.—The water-level in the boiler is shown by means of a glass gauge fitted between two cocks, the upper one of which is in communication with the steam space and the lower with the water space of the boiler. The gauge glass is generally made

in the form of a tube, as shown in Fig. 84, about $\frac{5}{8}$ inch external diameter, and about 14 to 18 inches long; but as pressures have increased, other methods of fitting the glass are also adopted. The gauge for tank boilers is preferably attached to a steady pipe (see Fig. 82) with large openings, not less than 2 inches in diameter, so that there is no danger of their becoming choked, and thus showing a false height of water. Care should be taken not to make the

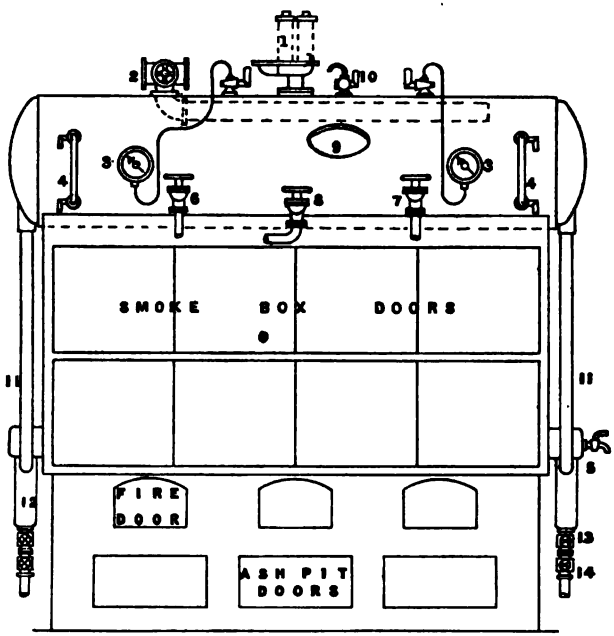


FIG. 83.—Mountings on Water-tube Boiler.

connections with the boiler near any current of either steam or water, because the level may be affected by a slight difference of pressure; for each $\frac{1}{10}$ lb. the level is altered by about $2\frac{1}{4}$ inches. All boilers should have two sets of water gauge mountings, and double-ended boilers are generally fitted with three. The lowest water-level in tank boilers, shown by the gauge, should be at least 4 inches *above* the highest point of the heating surface.

Three cocks are fitted, as shown in Figs. 84 and 86, to each water gauge: the top, or steam, cock connects the top of the glass gauge with the steam space; the bottom, or water, cock connects

with the water space; and the drain cock is fitted below and opens connection when required with the bilge. The gauge should be tested and blown through occasionally. To clear the steam cock, open the drain and close the water cock; steam should then blow freely through the drain, and the water-level disappear from the gauge. To clear the water cock, shut the steam cock and open the drain; water should come freely through the drain, but the water-level may not be affected in the gauge; most probably the level will rise from condensation in the upper part of the tube. In normal working the top and bottom cocks are open and the drain is closed; all the handles should hang downwards when in their normal working position, and there is then less probability of derangement.

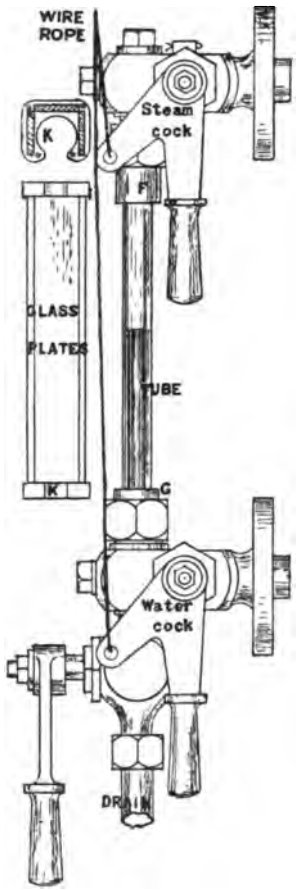


FIG. 84.—Water Gauge.

The glass tubes used for water gauges, and other parts subjected to steam and water pressure, should be fire finished, and not *cut* to the required length; cutting reduces their durability.

Water Gauge Safety Fittings.—The ordinary glass tube fractures very easily, and to protect the men in the stokehold from splinters, a glass protector is frequently fitted outside the tube. In Fig. 84 the protector is shown removed to the left of the gauge. It consists of two or three sheets of plate glass, about $\frac{1}{2}$ inch thick, socketed into brass end plates, *K*, *K*, which are kept secured by long brass wires screwed and nutted as required. A section through the protector is also shown in the figure, and it will be noticed that each end plate is cut away for part of its circumference, so that it can be passed over the glass tube, then slipped upwards on the projecting part *F*, below the gland nut, and dropped down over the bottom projection; when the protector is in place, it safely remains until lifted off. The uncovered side of the gauge is usually towards the boiler, away from the stoking platform.

When a tube fractures, the steam escapes through the top, and water through the bottom connection with the boiler. Handles are fitted for closing the cocks from the stokehold platform, the connections for which are shown in the figure. It is preferable to also fit wire ropes for this purpose, and lead them away over rollers to some safe place. When wires are fitted, each cock can be opened or closed independently, but when there is a rigid connection between the two cocks they must be worked together. When steam only is escaping the cocks are generally accessible; but the escape of water is more dangerous, and in some cases an automatic arrangement is fitted to close the lower end of the tube. A small metal ball is supported on a ladle just below the end of the tube, and in such a position that any sudden escape, such as would be caused by a fracture of the glass, throws the ball against the end of the tube and closes the exit; once in position, the pressure inside the boiler holds it there. A similar arrangement is sometimes fitted to the top connection, but is not so necessary, for the reason already stated.

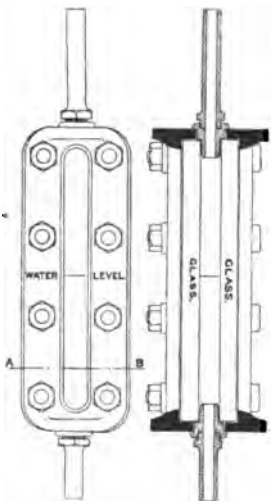


FIG. 85.—Window Water Gauge.

The glass tube used for containing the water and steam can only be made of a certain strength. If made very thick, the unequal expansion of the interior and exterior surfaces, especially in a strong draught, is very liable to produce fracture, and also the water-level is seen with difficulty.

Window or Plate Water Gauges.—For the high pressures now used, one or both of the usual water gauges, fitted to each boiler, are fitted with glass plates. The arrangement of cocks and steady pipes is the same, but instead of a glass tube, two flat plates of either very thin **talc** or thick **plate glass**, as shown in Fig. 85, are used to show the water-level. The plates are secured in place by two slotted brass covering plates, front and back, pressed on to packing against a central slotted piece in connection with the mountings used for the top and bottom cocks. The lamp, as in all cases except *reflecting*

gauges, should be placed *immediately behind* the gauge; but even then the water-level is not shown very clearly through the ruby-

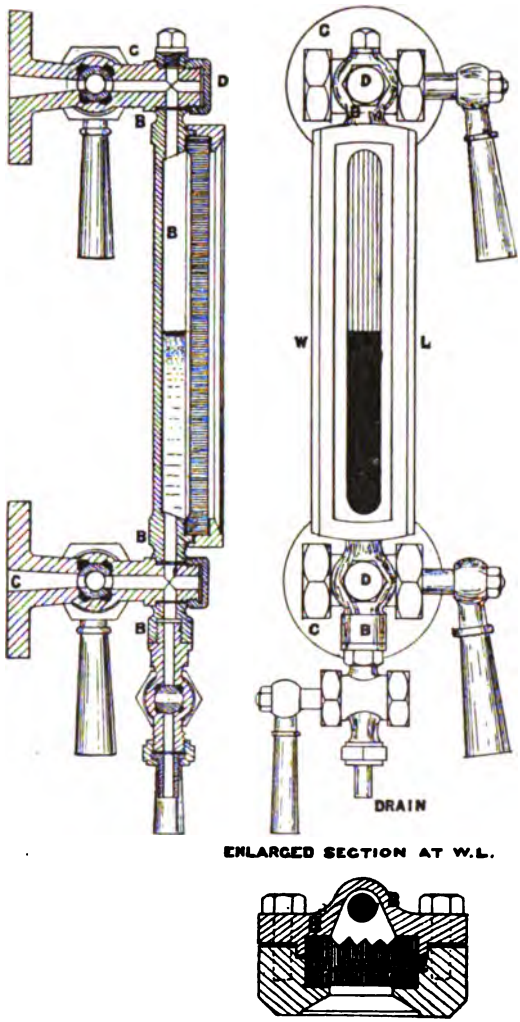


FIG. 86.—Reflex Water Gauge.

tinted talc. The glass plates are about $\frac{7}{16}$-inch thick, the talc about $\frac{1}{32}$-inch.

Klinger Reflex Water Gauge.—This type, shown in Fig. 86, is frequently fitted, and is suitable for any pressure. Like all

plate water gauges, there are no glands such as are required for packing the ends of a glass tube; the packing has been a fruitful source of accident by choking, or partially choking, the ends of the tube. The main body *B* of the Klinger gauge, which forms with the glass front plate the tube of the gauge, is bolted directly on to the top and bottom coned connections *C*, and secured in place by two capped nuts, *D*; the nuts, which are screwed with a *left-hand* thread, can be removed to clear the holes in the end connections. A gauge can be renewed in a few moments, but this is seldom necessary; Klingerite packing is used for making the joints. By reference to the enlarged section of the gauge, taken horizontally, it will be noticed that only one glass plate is used. The front of this plate is a plane surface, but the back, next the steam and water, is serrated in vertical lines of prismatic form.

When the light enters the glass *above* the water-level, there is no loss from absorption or transmission; it is totally reflected from the prismatic back, and shows up with brilliancy like silver. When the light enters the glass *below* the water-level, the rays do not emerge, but are refracted and absorbed, and consequently this part of the gauge shows up *dark*. The water-level is more clearly defined when the glass is clean, and a brush is supplied for the purpose; the top and bottom cocks are closed, and the top stud removed, when the brush can be passed through the tube to clean the inuer side of the glass. To prevent corrosion and eating away of the serration edges, the gauge should be drained and shut off when *not* in use.

Test Cocks.—Two or three asbestos-packed cocks are fitted at any convenient part of the boiler for testing the accuracy of the water-level. They are generally connected with the steady pipes, if fitted, and one is placed at a level about 2 inches above the highest point of the heating surface, and another about 10 inches above the first. If a third cock is fitted, it is placed about midway between the two.

When a steady pipe is not fitted, and for water-tube boilers generally, the cocks are placed about 5 inches above and below the working level, with a third cock, if fitted, midway between them.

Test cocks are not a very reliable guide, especially with high pressures; but when both water gauges are out of order they are better than nothing for testing the working height of a boiler.

Pressure Gauge.—This instrument is used to show the pressure

of steam in the boiler. The front face is graduated to a certain scale, and the pointer indicates the pressure at all times, as shown in Fig. 87. The gauge is fixed at a convenient place, well in sight and near the boiler with which it is in connection; two gauges are fitted to each boiler, one graduated to show about $1\frac{3}{4}$ times, and the other to show about $1\frac{1}{4}$ times, the working pressure.

A pipe, with an asbestos-packed cock or valve at the end nearest the boiler, connects up each gauge with the steam space near its highest part. Another cock is fitted in the connection with the gauge, and can also be used to drain away any water that

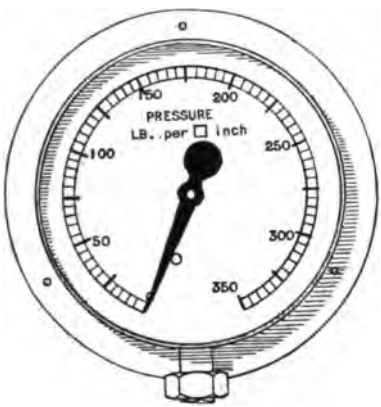


FIG. 87.—Pressure Gauge.

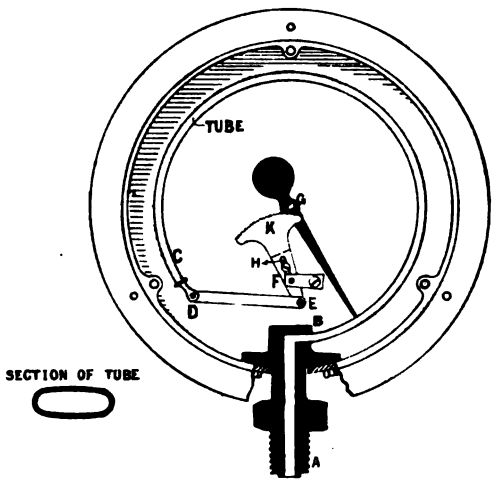


FIG. 87A.—Pressure Gauge Mechanism.

may accumulate in the pipe. It is generally arranged either that the drain must be open, or that the gauge is in connection through the cock with the boiler; in other words, that the gauge cannot be entirely shut off without some indication from the drain. The cock next the boiler is generally kept open by some safe means, so that it cannot be shut by accident or neglect.

The mechanism, shown in Fig. 87A, consists of a bent tube, *BC*, generally of oval section, which is internally connected with the external pipe *A*, and thus with the steam in the boiler. The end, *B*, of the tube is fixed. The other end, *C*, is connected by a rod, *DE*, with the short end of a lever, *EFK*, which has a quadrant, *K*, at its other end. The cogs or teeth on the quadrant engage in a pinion

wheel, *G*, which is attached to the spindle actuating the pointer. The bent tube under an internal pressure greater than that of the atmosphere tends to straighten itself; this is due to the greater length and area of the outer part of the bend of the tube, which produces a greater *total* pressure outwards. If the internal pressure is less than that of the atmosphere, the pressure of the atmosphere tends to bend it in still further, so that the gauge can register pressures both above and below that of the atmosphere. The proportions of the lever *EFK* can be slightly varied by means of the slot and screw shown at *H*.

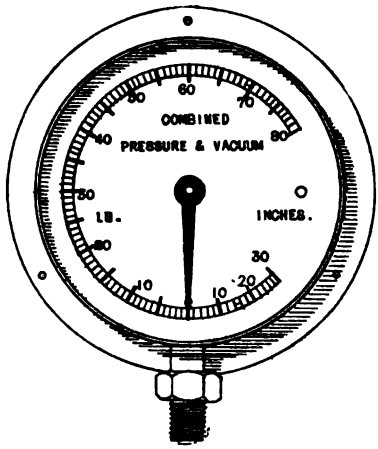


FIG. 88.—Combined Pressure and Vacuum Gauge.

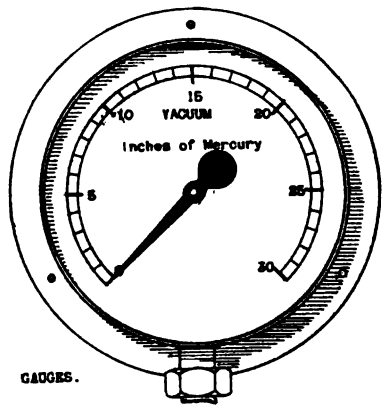


FIG. 89.—Vacuum Gauge.

The *zero* of the pressure gauge is that of normal atmospheric pressure (14·7 lb. per 1 square inch), and a stop pin is fitted to prevent the pointer falling below this point, or being carried right round the gauge, when it would indicate a much lower pressure than that actually present.

Combined Pressure and Vacuum Gauge.—For certain purposes, but not for boilers, a gauge which shows pressures both above and below the atmosphere is used. The zero point is still that of atmospheric pressure; pressures above are shown in pounds per square inch, but the vacuum is shown in inches corresponding with the mercury barometer, as shown in Fig. 88. These gauges are used for the engine receiver pipes, backs of slide valves, condensers, and, particularly, evaporators.

Vacuum Gauge.—The vacuum gauge (Fig. 89) is used to show only pressure below that of the atmosphere; the zero shown is that of atmospheric pressure as on other gauges. At 30 inches of vacuum, as shown by the gauge, the pressure corresponds to 14·7 lb. per square inch *below* atmospheric. In round numbers, 2 inches of vacuum represent an absence of 1 lb. pressure below atmospheric, and so on for each 2 inches; thus a vacuum in the condenser of 26 inches means that the pressure is about 13 lb. below atmospheric or the zero of the pressure gauge.

The internal mechanism is similar for pressure, combined pressure and vacuum, and vacuum gauges, as shown in Fig. 87A.

Air-pressure Gauge.—This instrument is used to show, in inches of water, the difference of pressure in two places, and is a form of water barometer. It consists of a bent glass tube of U-shape, as shown in Fig. 90, one end of which is connected internally with the outer air, and the other is open to the stokehold or connected by tubing with the place required. Coloured water is run into the tube to a convenient height, and when both ends are subject to the same pressure, the level of the water is the same in each leg. When from any cause there is a preponderance of pressure, as in a closed-stokehold, the water-level is depressed on that side, and there is a corresponding rise on the other. The difference of level which can be measured by the movable scale shown is the difference of pressure measured in inches of water.

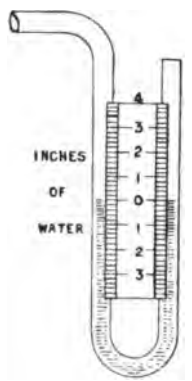


FIG. 90.—Air-pressure Gauge.

If, on the other hand, one end is connected with the interior of the funnel when steaming, and the other with the atmosphere, the water is depressed on the side open to the atmosphere, because the pressure in the funnel is less, due to the lighter products of combustion which have displaced the atmosphere. The difference of pressure is still registered, but as it is less than atmospheric, it can be termed vacuum.

The height of the water barometer is about 33 feet, and that of mercury about 30 inches, consequently the latter is less suitable for small differences. The water barometer gives an accuracy at least twelve times as great; thus half an inch on the water gauge

would only show about $\frac{1}{25}$-inch on the mercury barometer, which is hardly perceptible.

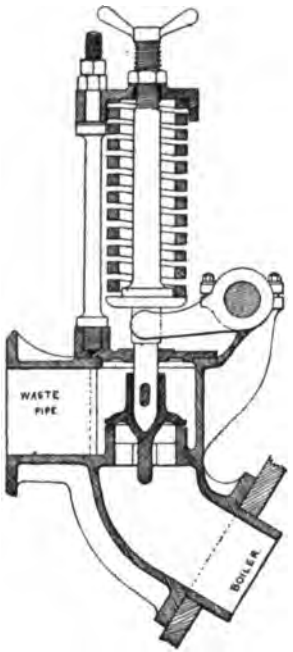


FIG. 91.

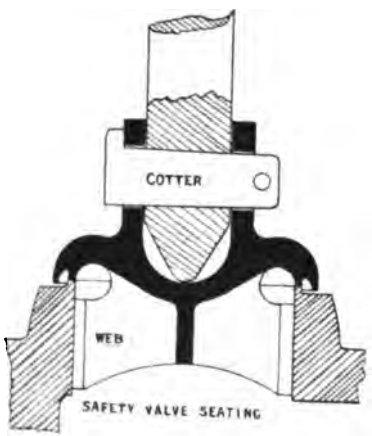


FIG. 92.

FIGS. 91 and 92.—Safety Valve and Seating.

Safety Valve.—In Fig. 91, one of a pair of safety valves is shown in sectional elevation as fitted to the marine boiler. It is fitted into the valve box, which is bolted to the boiler at or near the highest point of the steam space. For marine purposes the valve is loaded by springs of spiral form and square section, which are compressed the required amount, so that the valve lifts when the pressure inside the boiler is greater than the load on the springs. The valve is free to move on the spindle to a slight extent, so that if the spindle is not exactly in line with the valve, the latter remains firmly on its seating. In the sketch (Figs. 91 and 92) the valve is shown connected with the spindle by a cotter, which is a good fit in the spindle, but is allowed some slight play in the socket on the top of the valve. The pressure exerted by the springs is directed centrally on the valves. The top end of the spindle is fitted with a T-handle, so that the valve can be moved round on its seating. This is useful for removing grit or dirt by grinding it away from the face. When such movement is required, and the valve is under a high pressure from the springs, which is not compensated by the pressure inside the boiler, it is usual to partially relieve the pressure thus exerted on the valve face by using

the *safety-valve lifting gear* outlined in Fig. 82. A flange or collar is formed on the spindle to receive the spring; under this flange a forked lever is fitted, which is actuated by the lifting gear, the shaft of which is shown in a bracket to the right of the spindle; the gear can be worked, either separately or together, from the stokehold and the deck above. The arms and rod are now generally forged in one piece, and not connected by a key, as the latter may possibly work out of its place or tear away.

The *area* through the safety-valve openings is about nine-tenths of a square inch per square feet of heating surface; but the exact amount is laid down in the Board of Trade and other rules, and varies according to the pressure and other conditions of working for which the boiler is intended. The *lift* of the valves allowed is one-fourth the diameter, which gives the full opening due to its diameter; but it is seldom that the valve lifts more than about one-sixteenth of an inch even when all the steam that can be generated is escaping. Ferrules are fitted above collars on the standards supporting the springs to facilitate adjustment, and to prevent the securing plate being screwed down, and so load the springs above the proper working pressure.

When the boiler is tested above the working pressure the valves are *gagged* by inserting a ferrule below the securing plate, in place of the springs, which are temporarily removed.

At least two valves are fitted to each boiler, and frequently three or four are fitted to very large boilers. For very high pressures, when leakage is likely to occur from deformation of the valve, the diameter is limited to about $3\frac{1}{2}$ inches for 300 lb. working pressure, and to about $4\frac{1}{2}$ inches for 155 lb. Flat seatings are always preferable, and, whether flat or conical, the width of seating should not exceed one-sixteenth of an inch. A sketch is shown in Fig. 92 of a valve and seating used for high-pressure boilers. The cavity surrounding the valve seating is fitted to obtain, by reaction of the escaping steam, a definite opening of the valve and prompt relief of accumulation of pressure, not gradual leakage.

The safety-valve box should be entirely separate from the steam valves and internal steam pipe; in the case of some water-tube boilers, however, the safety-valve box is incorporated with the steam-valve box. A *drain pipe* is usually fitted so as to keep the valve box clear of water above the valves, and if a cock is also fitted, it should be kept open when fires are alight in the boiler.

The drain water is led into an open-topped drip tank, and is conveyed away by a pipe connected with the steam drainage system.

Steam Valve.—The steam in the boiler communicates with the steam pipe leading to the engines by means of a stop valve, which is usually self-closing; that is, when the pressure inside the boiler is suddenly released, and is then less than that in the pipe, the valve should close owing to the difference of pressure. By the above means the collapse of one of a group of boilers is isolated

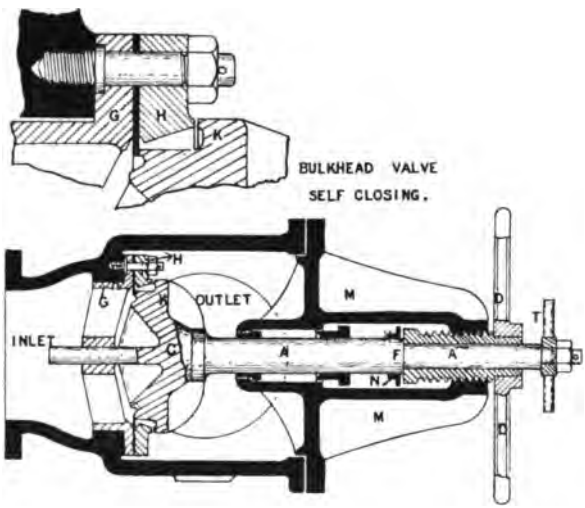


FIG. 93.—Self-closing Steam Valve.

from the remainder, and only the steam contained in it is allowed to escape through the fracture. A self-closing valve is shown in Fig. 93; the spindle *A* is horizontal, and made in one piece with the valve, so that it can be turned on its seating by the external T-handle. The amount of opening is regulated by the wheel *D* and hollow screwed spindle; and after setting the opening, which is generally shown by an index *N*, the valve must be pulled back against the collar *F* by means of the inner spindle and T-handle. When the pressure is greater inside the boiler than in the steam pipe, the valve opens itself, as it is then pressed towards the collar; but if the resultant pressure is in the opposite direction the valve remains closed. Great care should be taken that no undue force

is used to open it, and that when it is supposed to be open the spindle is actually withdrawn to butt against the collar. The packing of the gland spindle is important, and generally alternate layers of elastic core and asbestos are used, well smeared with mineral oil and black-lead; but in some cases other packings of special character are used. The area of opening through the valve is generally about one-fourth greater than the steam pipe used for carrying away the steam.

When the pressure is suddenly relieved in front of the valve—by fracture of the boiler or other cause—the valve closes automatically, and thus prevents the escape of steam from the pipes in connection with the back of the valve.

The ordinary type of flat valve-seating is generally used for boiler stop-valves; but for high pressures for *bulkhead* and other steam *valves*, a special construction, invented by Messrs Bevis and Gibson, is sometimes used with two seatings, the inner of which is flexible, as shown in the figure. The inner seating, when the valve is closed, butts on a flexible metal ring and makes a surface joint with it. The flexible ring is held between the flange G and the ring H. H is suitably made for the seating K of the back valve face, which is of the ordinary flat seated type, and, being solid, this seating prevents the valve being driven home too far on the flexible seating and so distorting it. Because the pressure always tends to press the flexible seating against the valve face, steam tightness is obtained with proper fitting.

Internal Steam Pipe.—The stop-valve box is generally made with a spigot, projecting into the boiler through the hole near which its flange is bolted, with which the internal steam pipe is connected. This pipe is generally made of gun metal, and runs horizontally along the upper part of the steam space, as shown in Fig. 83; slits extending about one-third of the circumference are cut on its top side, and the total passage area is slightly in excess of that through the valve. The object of the internal steam pipe is to prevent the passage of large bodies of water into the external steam pipe, and so to the engines. It acts as a separator inside the boiler; it remits the danger of violent priming, and generally acts as a preventative. For the Belleville boiler the pipe is pierced with a small hole only for each element, and this system is sometimes used for other types. Whenever the boiler is opened for examination, or cleaning, the openings are inspected to see that

they are clear, and, if necessary, the pipe is removed for the purpose.

An auxiliary steam valve is generally fitted of the same construction as the main valve, and is occasionally fitted to take steam from the main internal steam pipe; but a separate fitting is preferable, and more usual. The Belleville boiler is not usually fitted with an auxiliary valve, as the boiler is a comparatively small unit in the total power, and such a valve adds materially to the complexity and number of fittings.

Boilers which can generate great power are frequently fitted with *twin* internal steam pipes, which join into a single pipe where the connection is made with the spigot on the valve box.

Prevention of Priming.—If reducing valves are not fitted, the best method of preventing priming is to throttle the steam. This can be done at the boiler itself or at the bulkhead valve, or preferably at both places. An allowance for the velocity of the steam through the boiler stop valve of about 14,000 to 15,000 feet per *minute* approximates very closely to that obtained in practice. The opening of the valve should generally be limited to the amount required to give this velocity, and generally no priming will then take place. The reducing valve, even when fitted to each boiler, does not limit the output of the boiler. This arrangement of throttling is in this respect superior, but the faces of the valves and seatings are sometimes cut by the steam when very high pressures are used.

Air Cock.—An asbestos-packed cock is fitted at the highest part of the boiler for the exit of any vapour or air either when the boiler is being lighted up, or being filled to the crown with water for preservation when not required for steaming. Care should be taken that the cock cannot open from vibration—a catch is usually fitted for this purpose—and that it is closed immediately steam begins to blow through it, because it may otherwise become inaccessible.

Asbestos-packed Cocks.—For naval purposes valves are always fitted, if possible, in preference to cocks; but where a full and unrestricted opening is required valves are not always suitable, and cocks are used for indicators, separator blow-downs, boiler blow-downs and brines, drains from cylinders and steam pipes, water and pressure gauges, water test cocks, hydrometer and air cocks. In the merchant service, cocks are also frequently used for regulating the boiler feed and other purposes.

When cocks are used, and are subject to steam or water pressure, they are generally asbestos packed, as shown in Figs. 84 and 94. In Fig. 94 the plug does not touch the metal surface, but moves on a non-metallic bearing of specially prepared asbestos packing, which is caulked into grooves in the shell and vulcanised. This packing is shown in black section, and in the shell by dotted lines. The cone-shaped plug tends to be ejected by the steam or other pressure to which it is subject; to balance this pressure a small hole is drilled through the top end, as shown. Apart from this balancing, the plug is held in place by a spring holding-down plate fitted to bear on a collar of the plug spindle. Below this plate a gland is fitted and packed to keep the spindle pressure-tight.

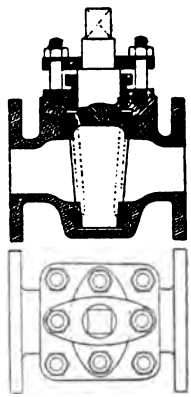


FIG. 94.—Asbestos-packed Cock.

For water gauge cocks and other parts where it is possible to fit them, it is preferable to fit the handle at the small end of the plug-cone, as shown in Figs. 84 and 86.

Feed Valves.—Usually two feed valves are fitted to each boiler, one in connection with the main and one with the auxiliary feed pump. When looking at the boiler, the main feed valve should be on the right and the auxiliary feed valve on the left of the boiler. The valves are always of the non-return type, that is, they close to pressure from the top or boiler side of the valve; the pressure below the valve must therefore always be in excess of that in the boiler, or the valve closes. A Belleville generator type of check and non-return valve and box, used for high pressures, is shown in Fig. 95.

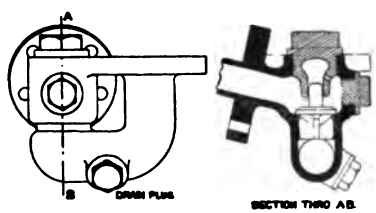


FIG. 95.—Feed Check Valve.

For ordinary hand regulation, gearing is fitted, if required, to work the valve from the stokehold platform. The valve seating is generally raised sufficiently to facilitate repair. Valves with flat seatings reduce hammering, and do not easily become fixed in the openings, and for high pressures they are usually fitted in preference to coned seatings. In Belleville feed valves (Fig. 95), the valve and seating are both coned, but at different

angles, so that the upper edge of the valve cone butts on the seating.

The internal feed pipe is connected with a spigot on the valve box, and the water should be directed through it in the same direction as a descending current, which should also surround it, and prevent steam being formed in the pipe. A small hole is generally drilled near any possible pocket or bend, to allow any steam or air to escape; this reduces the stress on the pipe and work on the pump.

Automatic Feed Regulators.—An automatic feed regulator is fitted to nearly all water-tube boilers in addition to the ordinary valve described above. An automatic apparatus is unnecessary for tank boilers, because with the comparatively very large amount of water contained in them there is very little fluctuation of water level; whereas if the feed supply were shut off a destroyer's boiler for 10 to 15 minutes when under full steam, all the water would be boiled out of it.

Thornycroft Feed Regulator.—In Fig. 96 a part section of the steam chest of a water-tube boiler is shown, with a diagram of the arrangement of the Thornycroft automatic feed regulator.

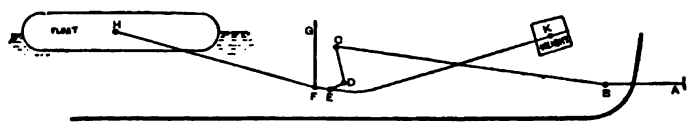


FIG. 96.—Thornycroft Feed Regulator.

A small wheel, *A*, which is shown in Fig. 48, and which can be worked from the stokehold platform, moves a screwed spindle in a bracket outside the boiler. The spindle passes into the interior of the steam chest, and is joined at *B* with a long rod, *BC*. This rod operates on a bell crank, *CDE*, pivoted at *D*, and by working the wheel the point *E* may be raised or lowered as required.

A float, *H*, which rises and falls with the water-level, and a balance weight, *K*, are connected by a bent lever, which is pivoted at *E*. The spindle *FG* of the internal feed valve is attached to this lever at *F* by a short connecting rod with ball and socket joints. The rise and fall of the float thus regulates the opening of the valve and the feed supply to the boiler. For varying the normal working height, the wheel and bell crank are used to raise or lower the point *E*, at which the float and weight are pivoted.

The valve is of the double-beat type, as shown in Fig. 97, and as no glands are fitted, a perfectly free movement with very small friction is obtained. The pressure in the feed pipes is generally about 50 lb. per square inch in excess of the boiler

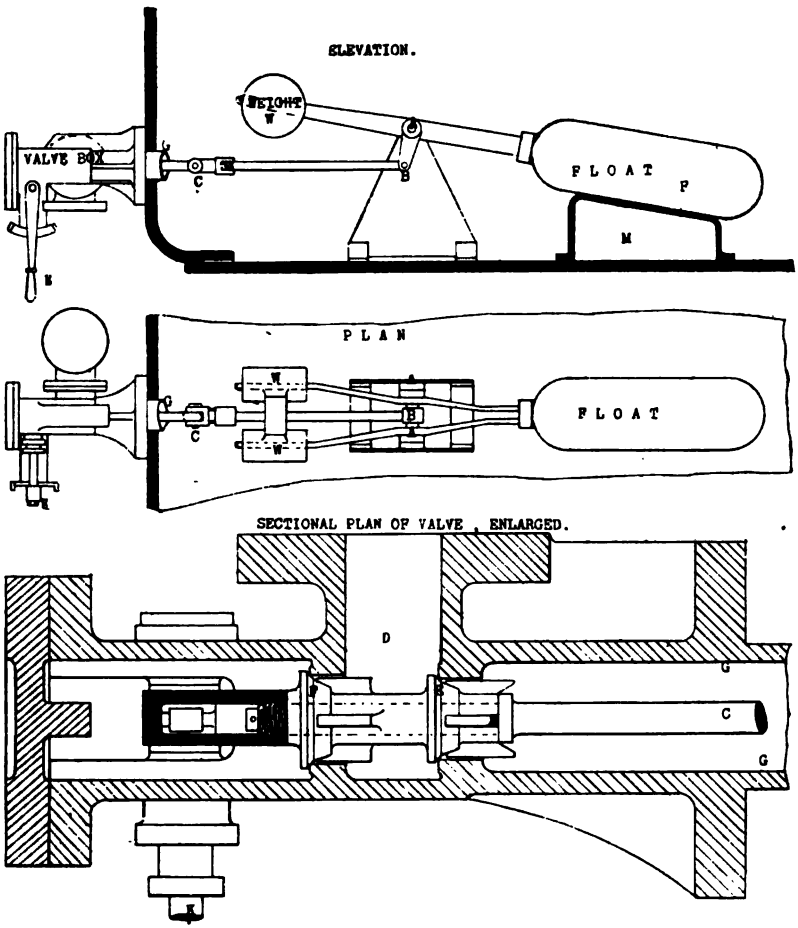


FIG. 97.—Bevis Feed Regulator.

working pressure. The usual valve is fitted outside the boiler, and hand regulation can be used.

Bevis (Laird) Feed Regulator.—In Fig. 97 the float *F* and balance weights *WW* are fitted inside the steam chest, and are pivoted at *A*. The movement of the float, following the rise and

fall of the water-level, works the lever *AB* and rod *BC*, attached to a double-beat valve, *EF*. The feed water enters through a non-return valve to the pipe *D*, and when the float falls and opens the valves *E* and *F*, which are attached to the same spindle, and therefore work as one piece, the water passes through the pipe *G* into the boiler. There is an open connection between the space at the back of the valve *F* and the pipe *G*. A hand lever, *K*, is fitted for testing the free movement of the valve and its connections, and it is possible to regulate the opening of the valves, independently of the float gear, by setting the lever in fixed positions on the arc shown.

The pipe *G* is sometimes fitted with an internal fork, to direct the flow of the feed water by separate pipes, either directly into each of the two downcomers, or to distribute it inside the water space. A rest, *M*, is provided for the float inside the steam chest, and vertical guides are fitted between the longitudinal boiler stays, to prevent the float being thrown from side to side, and to steady its movements. The valves are of special coned shape below the seatings, so that the opening is gradual, and this produces a tendency to a continuous flow, and not an intermittent one.

The advantages of this arrangement are that—

1. When steam is not up, the valves are accessible without opening the boiler.

2. The float arrangement can be thrown out of gear, and hand regulating adopted, without disturbing any of its parts.

3. There are no internal feed pipes under pressure.

The float is subjected to external pressure, and is entirely closed. Its weight, when empty, is tested, and after being subject to an external test pressure for some considerable time, it is again tested, and, by comparing its new weight, any leakage can be detected.

White-Forster Feed Regulator.—A sectional elevation of the regulator and an outside elevation are shown in Fig. 98. The float chamber being in connection, through steam and water pipes, with the boiler, the float rises and falls with the water-level in the boiler. Steam being allowed free access, through the top, to the interior of the float, the internal and external pressures on the float are balanced. The float can therefore be made of the lightest material, and consequently requires no weight to balance it.

The movement of the float, moving the lever *FXC*, pivoted at *C*, and the rod *D*, works the valve *V*, which moves inside a double piston, *P*. The piston *P*, shown in black section, copies the

movement of the valve *V*, and being connected by a rod, *R*, with the external lever *LRM*, regulates the opening of the feed cock *K*. The piston is made of two areas, the larger above and the smaller below. The boiler pressure inside the float chamber, always acting on the lower end, tends to force the piston upwards. The steam

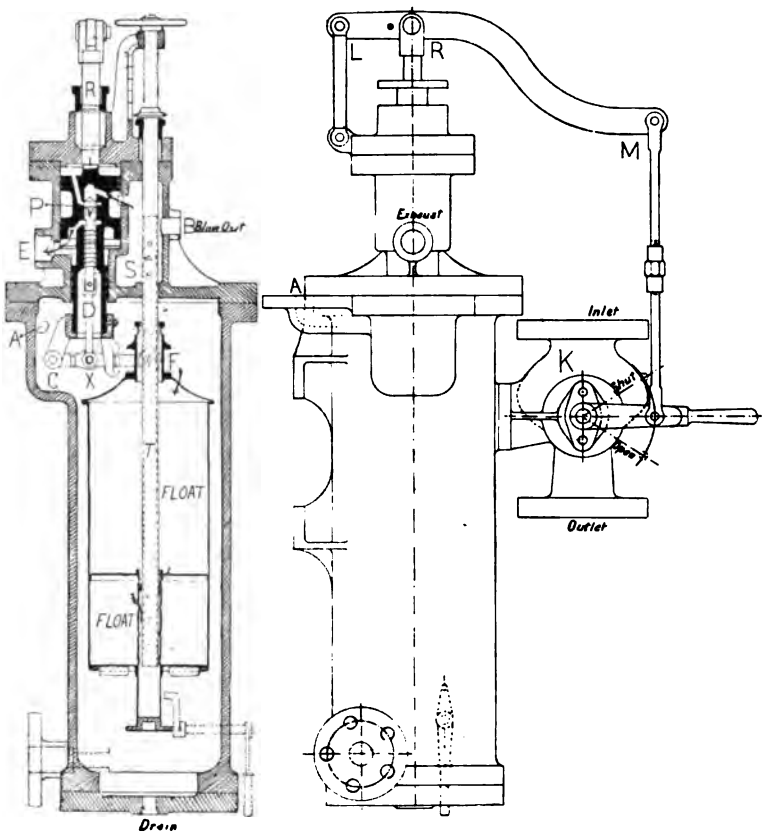


FIG. 98.—W.-F. Feed Regulator.

required for working the upper piston passes downwards from the float chamber through the float, and upwards from the bottom, through an internal pipe, *T*, into the steam space *S*. The exhaust steam passes either into the waste steam pipe or into the auxiliary exhaust.

When the water-level falls the valve *V* is moved downwards, and admits steam from the space *S* to the space above the larger

piston. The total pressure on the larger piston overcomes that on the lower piston, and, pulling the external lever downwards, opens the feed cock *K*.

When the water-level rises the valve *V* is moved upwards, and releases the pressure above the larger piston into the exhaust. The constant steam pressure below the lower piston overcomes the *exhaust* pressure on the larger piston, and, pushing the external lever upwards, closes, or tends to close, the feed cock *K*.

The normal working level in the boiler can be adjusted for a few inches, more or less, by allowing a certain depth and weight of water to accumulate inside the float, and thus increase its virtual weight. The increased weight of the float requires an increased depth of water to float it at the original level, and therefore the water-level in the boiler must be correspondingly raised.

Thus, *to raise the normal working level in the boiler*, the steam connection *A* with the float chamber is closed, and the blow-down *B* opened, until sufficient water accumulates inside the chamber to overflow into the float. The tube *T* is then raised the desired amount by the thumb screw, as shown by the index, when the steam connection is again opened to the boiler, and the blow-down is closed as soon as dry steam is emitted from it.

To lower the normal working level in the boiler, the tube is lowered the desired amount, and the blow-down opened until dry steam is emitted from it.

A hand lever is attached to the cock, so that the free movement of the lever and piston can be tested; and another lever is fitted, at the lower part of the float chamber, for testing the free movement of the float and its connections.

The advantages of this feed arrangement are—

1. Compactness, and visible action on the feed cock.
2. Lightness.
3. All its parts can be made of brass or gun-metal, and are not subject to corrosion.
4. The float action is entirely free and unencumbered by stuffing boxes.
5. All the working parts are attached to the top cover, and, by shutting off the connections with the boiler, can be examined when under steam.
6. The feed can be regulated by hand, if desired, and the float can be easily put out of and into gear.

Mumford Feed Regulator.—As shown in Fig. 99, there is a float and float feed regulating box fitted at about the working water-level outside the boiler. The float box is connected through two straightway cocks with the steam and water spaces. Below the float box and at any convenient position is the feed check valve, which can be used either for hand or automatic regulation. The check valve is of the type which is balanced by fitting a piston on

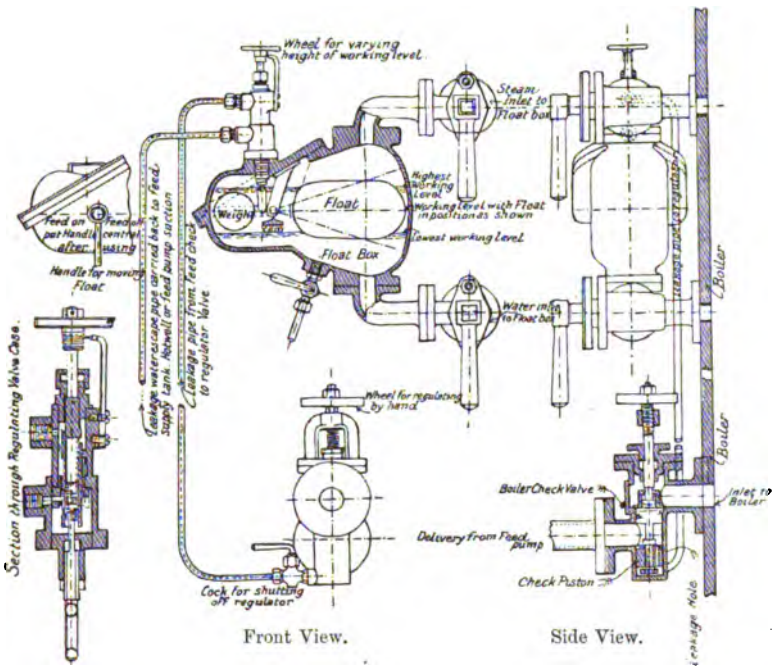


FIG. 99.—Mumford Feed Regulator.

the spindle below the valve, and the boiler pressure above the valve tends to keep the valve closed at all times. There is a small hole through the piston which tends to equalise the pressure above and below it, and in connection with the space below the piston is a leakage pipe leading to the regulating valve which is operated by the float. When the float rises with a rise of water-level it opens the regulating valve, and, releasing the pressure below the piston, causes the check valve to close. When the float falls with a fall of water-level it closes the regulating valve, and, preventing leakage

from below the piston, allows the pressure below the check valve to open it against the pressure in the boiler.

The advantages of the Mumford system are that: one size of float regulator and valve can be fitted to any size of boiler, and is independent of the bore of the feed pipe and check valve; all parts are made of gun metal, and therefore not subject or conducive to galvanic action; the working parts are compact, of light weight, and sensitive, and, being fitted outside the boiler, can be easily examined at any time without opening the boiler; a hand regulation can be resorted to by simply closing a cock on the leakage pipe.

Brine Valve.—On nearly all boilers a valve is fitted at some convenient part, and is connected, through a spigot, with an internal pipe, which is fitted with a scum pan slightly below the working water-level. The other side of the valve is connected, through a separate pipe and an asbestos-packed cock, with the sea valve. The brine or surface blow-down is used to remove any floating grease or other matter from the surface of the water in the boiler. The matter thus removed must pass through the brine valve on the boiler, the brine cock, and the sea valve, which may also serve for a group of boilers and for the blow-down. The valve on the boiler is fitted so that the pressure tends to keep it closed; this is used also for feed and blow-down valves.

The brine and blow-down cocks are fitted below the stoking platform, and a handle is provided for each, which cannot be unshipped without closing the cock.

Blow-down Valve.—The use of the blow-down is to remove any sediment or solid matter from the water by blowing it through the valve, a separate pipe and a blow-down cock, and the sea connection valve, which is also used for the brine in many cases. The quantity of water, and of the heat thus lost, are given in the chapter on "Evaporators."

For water-tube boilers the blow-down is fitted in addition to a drain valve, with which a hose can be connected for running the water into the bilge, the reserve tanks, or into another boiler by levelling. Inside the boiler the mouth of the pipe which is connected with the drain valve is frequently fitted about a foot or so above the bottom, so that only clean water is thus run off.

Hydrometer or Salinometer.—Sea-water contains by weight about $\frac{1}{32}$ part of solid matter, and if a certain portion of the water

be evaporated, the proportion of solid matter remaining is increased. Thus, if two gallons of sea-water are boiled until only one remains, the density or proportion of solid matter will be double that of the original sea-water; in other words, it will contain $\frac{2}{32}$ parts by weight of solid matter. This proportion, or density as it is usually called, is measured by the hydrometer or salinometer, which consists of a weight, float, and graduated stem, as shown in Fig. 100. The density is indicated, on a scale, by the graduation at which the hydrometer floats in the liquid. In the mercantile marine the graduations are sometimes shown in thirty-seconds, thus $\frac{1}{32}$, $\frac{2}{32}$, $\frac{3}{32}$, and so on; but for naval purposes each thirty-second is divided into 10°—thus, the density by the hydrometer of fresh water is *zero*, of sea-water it is 10°, of twice the density of sea-water it is 20°, and so on.

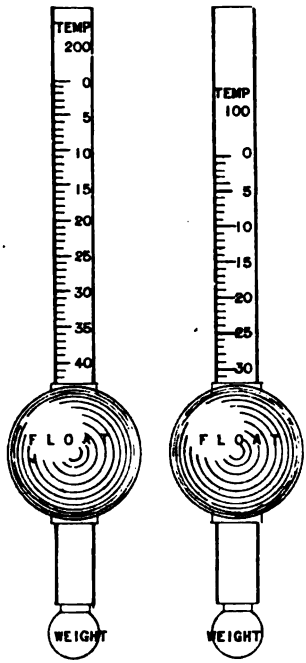


FIG. 100.—Navy Hydrometer.

The density of the water tested depends also on its temperature, and, consequently, the graduations are made to suit the temperature for which it is used. The ordinary naval hydrometer is graduated for two temperatures—that for 100° Fahr. is shown on one side of the stem, and that for 200° is shown on the other, and the temperature is shown near each scale. The two sides are shown in the figure, and it will be noticed that there is a difference of about 10° between them, due to temperature only.

When the scale is graduated for a temperature of 100° Fahr. it is useful for testing the feed condensed water, and when graduated for 200° Fahr. it is useful for testing the water drawn from a boiler under steam. When great accuracy is required a thermometer should be used for ascertaining the temperature at the time of testing, and the water should be free from steam and vapour.

The boiling-point is also affected by the density of the water, and for each $\frac{1}{32}$ part, or 10°, of solid matter it is about 1·2° Fahr.

higher. Thus the boiling-point of ordinary sea-water is 213·2° instead of 212°, which is the boiling-point of fresh water at atmospheric pressure.

Richards' Sensitive Hydrometer.—The comparatively very small amount of water contained in water-tube boilers is proportionately greatly increased in density by the same amount of solid matter pumped into the boiler, and the density may rise very quickly unless it is immediately detected. The ordinary hydrometer, such as described above, is not altogether suitable for the detection of slight increases, and a sensitive hydrometer is now used whose graduations are ten times longer, as shown in Fig. 101. The size of the float is increased and that of the graduated stem decreased, until the relative proportions of the two become suitable for this more minute detection; the principle is just the same in both types. This hydrometer is graduated for a temperature of 100° Fahr.

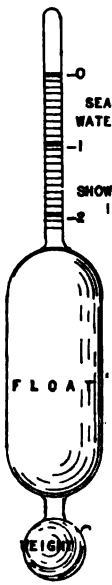


FIG. 101.—Sensitive Hydrometer.

Chemical Tests of Boiler Feed Water.—With the various testing apparatus supplied by chemists or engineering firms for use on board ship, full instructions are usually given, and it is only necessary here to describe how these tests are made in practice. In all chemical tests the water should be at a temperature of above 60° and below 100° Fahr.

Red litmus turns blue in the presence of an alkali; and blue litmus turns red or tends to bleach in the presence of an acid. Boiler water should always be alkaline and never acid, and in general the litmus shows no change either from red to blue or from blue to red, but a comparison will sometimes give a more definite result; thus, tear a selected piece of litmus in two parts, and immerse one piece and stir it about in the water under test for some minutes, and then compare any change of tint from the original for either acidity or alkalinity.

Phenolphthalein affords a more delicate test for alkalinity. One drop in 50 cubic centimetres, a large test tube, should turn the water to a magenta-pink colour if a sufficiency of lime is present and the water is alkaline; but if the phenolphthalein is a weak solution or weakened by exposure to air, two or three drops may be necessary to produce the same effect. If no change is effected it

denotes acidity, or tendency to acidity, according to the number of drops added and the strength of the solution.

The *silver nitrate test* for chlorine shows the presence or otherwise of salt in the feed water. One drop of the ordinary silver nitrate solution should produce a white cloudy precipitate in a small test tube, 25 cubic centimetres, of feed water if salt is present. The proportion of salt water present can be obtained by next adding (after the cloudy precipitate is formed) one drop of *potassium chromate*, when the cloudy precipitate changes to a yellow colour. If this yellow colour changes to red after adding another drop of silver nitrate solution, the proportion of sea-water present is almost inappreciable, and generally the feed water may be considered in a satisfactory condition; but if it is necessary to add several drops before obtaining the red colour, a search should be made for the cause, and it generally points to a leaky condenser. This test in experienced hands gives a definite result; for example, in an actual test with the usual naval stores and appliances, 25 drops of silver nitrate solution just produced a red colour when the density showed $\frac{1}{10}^{\circ}$ on the sensitive hydrometer, that is, when the proportion of sea-water present in the feed water was only 1 per cent.

Zinc Protectors.—Slabs of rolled zinc, generally $12 \times 6 \times \frac{1}{2}$ inch in thickness, are suspended inside the boiler at any part that is likely to suffer from corrosion. The slabs are connected with the stays by flat mild steel staves secured by bolts, with squares under the heads to prevent them turning in their holes and slacking off. The connections are carefully cleaned and filed bright, so as to obtain proper electrical contact between the stay and the slab.

Ash Shoot.—A circular tube, about 2 feet in diameter, is generally fitted from the level of the main or upper deck, down through the stokehold hatch to the stoking platform. This serves as a passage-way for the buckets of ashes hoisted from the stokehold, and they are then emptied overboard through a shoot on the ship's side. The tube also serves as a ventilator in many instances, and then it is continued upwards until it forms the seating for a cowl, which may be either built of light steel plates or made of canvas. The opening necessary for getting the bucket from the tube is closed by a door when not in use, and, to prevent accidents, it is only of the height necessary for extracting the bucket. At the lower end, a limited height from the stokehold plates prevents any one from getting below the shoot without warning.

When closed-stokehold forced-draught is fitted, it is necessary to close the tube to prevent a back draught through it; a moderately air-tight door is therefore fitted both below and above the tube openings, one of which is supposed to be kept closed when the other is open. With this system of forcing, the ash shoot is generally a separate fitting from the ventilating tubes, or only used in part when the fans are not working.

The ashes are hoisted by a small double-cylinder engine controlled by a differential reversing valve, or, as in the later ships, by an electrically driven hoist. The bucket, in either instance, is prevented from over-hoist by automatic stopping gear. A gong and voice tube are also generally fitted between the hoisting position and the stokehold.

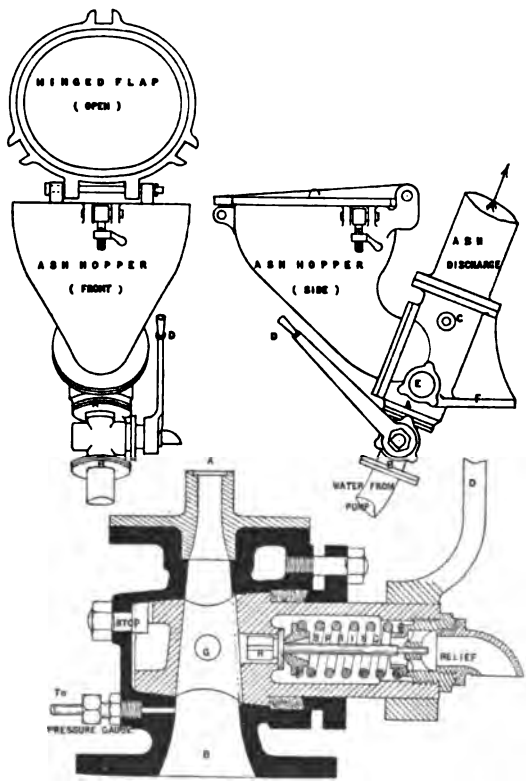


FIG. 102.—See's Ash Ejector.

See's Ash Ejector. — For continuous working at high rates of combustion the ordinary ash hoist is cumbersome, and an excessive number of men are continuously employed in getting up the ashes from below and throwing them overboard. With See's arrangement there is a saving in men, and the stokeholds, which are always too small, can be kept clear of ashes and accumulation prevented. The arrangement is shown in Fig. 102, and consists of a hopper (just above the stoking platform) into which the ashes are thrown. The hinged flap is closed at first starting.

Water under pressure from the fire-main is then admitted, through a pipe *B* and cock, by moving the handle *D*. The water carries the ashes with it through the discharge pipe, which leads up to the ship's side above the water line. A section (enlarged) through the valve is shown in the lower figure. A pressure-gauge is fitted to show the water-pressure, and also a relief valve in case the pressure gets too high when the shoot is choked. A clearing hole is fitted at *E*. If the discharge pipe is well proportioned and the bend is very gradual, good results are usually obtained, choking is seldom troublesome, and the ashes are thrown well clear of the ship's side.

Armour Gratings.—Nearly all warships are fitted with armour gratings, as a continuation of the armoured deck or decks, inside all ventilators, hatchways, and funnels, where water-tight doors cannot be fitted, to protect the parts below them as far as possible from shot and shell. In some instances armour hatches or sliding doors are fitted to close above the gratings, for use in extreme urgency; but all of them cannot be safely closed when men are below in machinery compartments, owing to the great heat and lack of sufficient ventilation.

The gratings are somewhat similarly arranged to the firegrate, but the bars are of deeper and rectangular section. A few of them in each hatch are bundled together, and hinged as a whole to form an opening when required. The gratings occupy some of the available area, and, in the interior of the funnel, a net area of opening must be allowed between the gratings for the escape of the funnel gases; the gratings increase the size of the base of the funnel, which is contracted gradually above them to the required net opening.

Fire Extinguishers.—These are fitted to boilers used for strong forced-draught in case a sudden loss of water in the boiler, or other accidental occurrence, necessitates putting the fires out. Pipes in connection with the sea, the level of which is always above that of the firegrate, are fitted in front of the boiler (and sometimes at the back also) leading to two or more openings through the casing. The control valve can be worked from some place outside the stokehold, and, when opened, water is sprayed on to the fire. The safety arrangements, such as self-closing fire and ashpit doors, prevent the steam generated in the furnace from gaining access to the stokehold, and the steam therefore finds its way up the funnel.

The fire extinguishers are tried periodically, and always immediately before starting a full-power trial. The position of these fittings is shown in Figs. 48 and 50. No fire extinguishers are fitted for boilers entirely fired by liquid fuel.

Sweeping and Searching Tubes.—The best method of external cleaning, so that the efficiency of the heating surface can be continuously maintained while under steam, is to provide for manual appliances for sweeping—using preferably a wire brush. The better classes of upright water-tube boilers are so constructed that a brush can be passed from the stoking platform, between consecutive rows of tubes, to the back end. The tubes are arranged in straight rows from front to back, and each row has approximately the same curvature. The circuitous route given to the gases in some boilers of the Normand type prevents the efficient sweeping of the tubes, and also appears to be attended with a higher coal consumption even when clean.

For searching and cleaning tubes internally the figure shown of the White-Forster boiler illustrates the best system. For the Yarrow straight tube boiler, and for other boilers with curved tubes, a flexible handle is necessary and is not so easily manipulated. A suitable length of wire rope is perhaps the best type of cleaning tool, and can be worked by one man only.

With many of the chain scrapers two men are required, and if the tube is not already fairly clear it cannot be cleared by this means. In 1895 the author used for boilers of the early Thornycroft type first a wire rope, and then washed through with water from the fire-main. For the latter purpose a length of pipe, about one inch in diameter, was connected with the fire-hose at one end, and at the other, with a right-angle bend, a nozzle was formed, so as to just enter the open ends of the tubes in the steam chest. One man manipulated the pipe from inside the steam chest, while another watched the bottom ends of the tubes to see that a full flow of water ran through each tube. This method was successful, and a boiler with nearly a thousand tubes could be completely searched in a few hours without a large expenditure of fresh water, which was always used for the purpose. The tubes are always searched periodically, and it is usual to search them before steaming at full power.

For boilers of the Babcock and Belleville type with small doors opposite the ends of the tubes, a scraper or wire brush is commonly

used for cleaning the interior. The doors are, of course, previously removed, and, after scraping, each tube must be washed through, because otherwise the scale would lodge in the nearly horizontal tubes and end-pockets. Although the number of tubes is less, and the room for working is greater, it takes about four times as long to open out, clean, and close up a Babcock boiler than it does one of the White-Forster type of about the same power.

Boiler Bearers and Anchoring.—A cylindrical boiler rests on two or more bearer frames built above the inner frames of the vessel, and extending from one-fourth to one-sixth around the shell circumference as shown in Fig. 103. Chocks are fitted at front and back to prevent movement of the boiler in these directions, and being secured to the ship, not to the boiler, thus allow sufficient expansion of the latter when under steam. The boiler is allowed free room for expansion circumferentially and upwards, but it is generally anchored loosely, when cold, to the bearers, by bands of flat steel plates, 3 to 6 inches wide, which are secured down to the bearers at B; horizontal stays are fitted near the top as shown.

Lagging.—Non-conducting material, generally of asbestos or silicate cotton, is fitted around the upper part of the shell of cylindrical boilers and over the back plates, and is carried down to a level just clear of the boiler bearers. The lagging is kept in place by thin sheet galvanised steel plates and angle-steels. The front of the boiler above the smoke boxes and behind the uptake is generally lagged in a similar way, and care should be taken that the uptake and air casing in connection with it should be fitted sufficiently clear of the boiler for periodical examination of the nuts and stay-ends, etc.

Air Casing and Uptake.—The smoke box of a cylindrical boiler is built up of thin plating and jointed on to the boiler, and is kept clear of the tube ends to allow freedom of withdrawal of all tubes without its removal. The lower and generally horizontal parts are provided with small doors, sliding in grooves, for the ready removal of soot when under steam, and when it is not advisable to open the smoke box doors for the purpose. In front of the casing and opposite the tube ends, doors are fitted, which when open allow the tubes to be swept when under steam, as shown in Fig. 103. The uptakes, shown in Figs. 103 and 104, should be as far as possible without sharp bends so that the gases may have a free exit to the funnel through it.

Around this inner casing, which forms the smoke box and uptake, there is always a second thin casing fixed at about 1 to 2

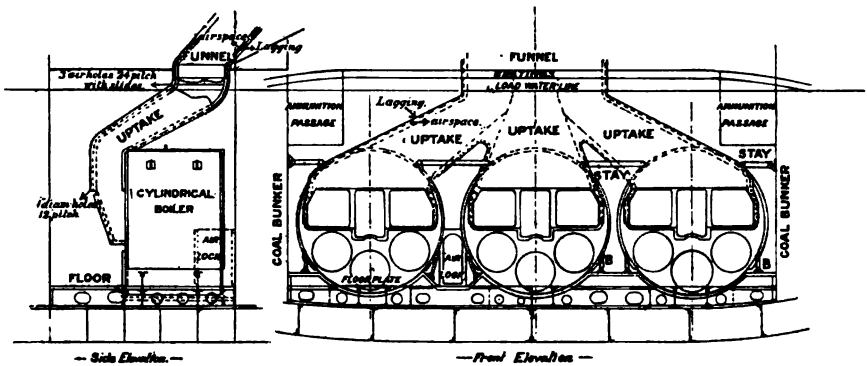


FIG. 103.—Cylindrical Boiler Uptakes.

inches from it. The air from the stokehold has free access to the space between the two casings, and is allowed to escape either into the stokehold at the highest point, or into a space formed by a

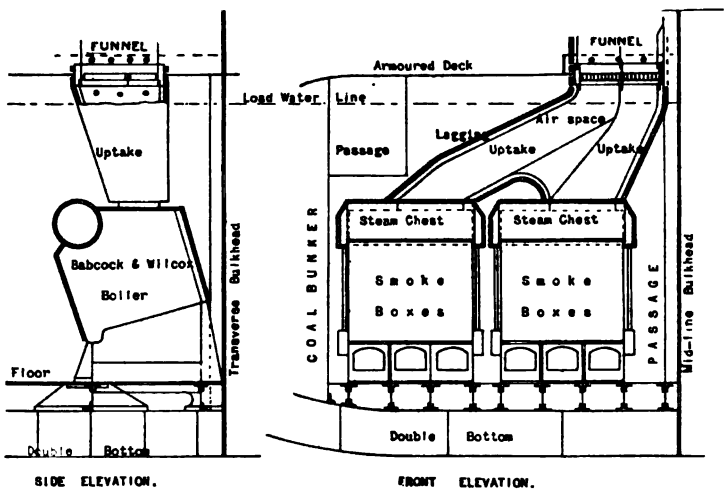


FIG. 104.—Water-tube Boiler Uptakes.

continuation of the second casing to some height above the upper deck and around the funnel.

The smoke-box doors are also fitted with a second casing in the

same way, but in this instance the hot air escapes immediately above the doors. Another thin system of plating (generally galvanised plates are used for all casings) is fitted across between the boilers and parallel to the front of them at about 18 to 24 inches back from the front end. This casing prevents the dust and ashes being carried into the space behind, and, with closed-stokehold forced-draught, assists in maintaining the air pressure near the furnace fronts. Doors for convenience of access to the parts behind are fitted as necessary. In some cases, the space so formed on each side of the boiler is utilised to warm the air which may be supplied to the fire near the back end of the grate. It may be seen in the illustrations of Belleville, Yarrow, and Thornycroft boilers, and is also used with other types.

Air-Supply and Brickwork.—In nearly all water-tube types air is admitted at the back of the fire as well as at the front in the usual way. The boilers, generally used for high rates of forcing and combustion, are fitted with self-closing doors on the ashpits, both at back and front. Arrangements are made (see Yarrow-Howden system on page 213) for keeping the doors at the back closed when required. The casings are generally made double, and the space so formed is utilised for heating some of the air supplied for combustion. In a few cases this warm air is supplied below as well as above the grate. The circulation of air through these spaces keeps the casings cool, and thus, in addition to promoting efficiency of combustion, it keeps the stokeholds cool. The air doors in the casing are shown in the various figures. The bursting of a tube, although merely a temporary disablement of a boiler, is sudden, and the necessity for automatically self-closing doors to the fires, ashpits, and air-supply is apparent.

Brickwork is necessary, to a more or less extent, in all water-tube boilers, and is employed principally to protect the casings surrounding those parts of the furnace which are not flanked by the tube heating surfaces. The weight of brickwork is considerable, and in some early cases exceeded that of the actual boiler without water. Recently a very light and durable brick, or fire tile, has been introduced, and is used for those parts (and for baffles) which are not subject to blows from the firing tools. The bricks or tiles are sometimes secured in place by inset bolts to the inner galvanised steel casing; but the brickwork should, however, be as far as possible self-supporting, and should never be entirely dependent on the bolted

attachment. Various systems are shown in the figures, and in the Belleville and Laird boilers it should be noticed that conduits are formed in the brickwork for the air-supply above the fire.

Funnel.—The smoke and hot gases pass into the smoke boxes from the boiler, and are conveyed through passages, called the uptakes, to the funnel, as shown in Figs. 103 and 104. The area of opening of the funnel is usually from one-seventh to one-tenth the area of the firegrate. The plates separating the several uptakes are carried a sufficient height up the funnel, generally as high as the armour deck gratings, to cause the gases to move in approximately the same direction when they mingle together, and thus avoid loss of draught from shock or confusion of currents.

The funnel is surrounded (see Fig. 105), generally throughout its entire length, by a similarly shaped structure, called the outer funnel; holes are provided at the bottom to admit air into the space provided between the two, and at the top a bonnet extends outwards from the top of the inner funnel and overlaps the outer, leaving a passage-way for the heated air, and preventing rain from obtaining access to the space between them. The two funnels are stayed together by long bolts or rivets with distance pieces.

An additional casing is fitted at the lower part of the funnel, forming another air space. The heat radiated from the funnel is thus again intercepted, and the ascending current of air is carried up into the atmosphere a sufficient height above the upper deck to prevent inconvenience to the people on board. This outer casing is also used as the outlet for ventilating the bunkers and some other parts of the ship.

Telescopic Funnel.—In ships fitted with sail as well as steam power, the funnel was sometimes made to lower when not required. The upper part, nearly one-half, could be lowered inside the lower part, or hoisted again by means of chains and pulleys, worked either through worm or pinion gearing.

Funnel Guys.—The funnels are generally long compared with their bases, and have a tendency to topple over when in a sea-way. To prevent this wire-rope stays are attached to a band fitted round them at about two-thirds of their height, and secured through right- and left-handed stretching screws to eyebolts at the ship's side, or other convenient part. These stays are sometimes eight in number, and are radiated from the funnel to give support in all directions. The funnel lengthens when heated, and before lighting

fires the guys should be slacked off a sufficient amount to allow for this extension.

Very long funnels have an additional band round the funnel, about 6 feet from the top, and a second set of guys is attached to it.

Funnel Covers.—To prevent the deposit of moisture on the interior of the funnel when not in use, covers are fitted above them. They should overlap a moderate amount, and at the same time sufficient space should be allowed for the egress of the smoke from an auxiliary fire or airing stoves. Thin sheet-iron or steel plates were frequently used; but canvas covers are now generally supplied, and are more convenient for shipping and unshipping. The cover is supported by crossed rods raised over the mouth of the funnel; the canvas is spread over them, and laced down to the edge of the funnel top. When using airing stoves a corner of the cover can be turned back. The covers are put in place from the interior of the funnel, and a platform is fitted for this and other purposes about 4 feet from the top, and access is obtained by iron ladders permanently fitted inside and leading up from the armour gratings. Doors are fitted in the several casings for access to the interior of the funnel and uptakes.

PART III

COMBUSTION

CHAPTER X

STOKING AND PRACTICAL CONSIDERATIONS

Introductory.—Combustion is the chemical combination of a combustible with oxygen at a sufficiently high temperature.

The oxygen is supplied by the surrounding atmosphere, and in certain cases, now becoming very usual, artificial means are employed to increase the supply. This process is adopted in most steamships, and is called forced draught.

Average samples of fuel vary considerably; they are approximately composed of the following chemical substances:—

	Carbon, per cent.	Hydrogen, per cent.	Oxygen, per cent.	Sulphur and Ash, per cent.	Heat, B.T.U.	Actual Evaporation, from and at 212°, in lb. of water.
Scotch coal	78·5	5·6	9·7	6·2	14,100	9·5
Newcastle (N.S.W.) coal ...	82·0	4·5	4·0	9·5	14,350	9·26
Newcastle (U.K.) coal ...	82·0	5·0	6·0	7·0	14,820	9·83
Welsh coal	84·0	5·0	4·0	7·0	14,858	10·17
Westport (N.Z.) coal ...	84·0	7·8	4·0	4·2	14,800	10·97
Patent fuel	84·0	5·0	3·0	8·0	15,000	11·00
Petroleum, crude Beaumont	84·6	10·9	2·87	1·63	19,081 (by calorimeter)	

Of these substances carbon and hydrogen form the combustibles, and by their union, chemically, at a high temperature with the

oxygen of the atmosphere combustion is effected. It is true that combustible substances will unite with oxygen at all temperatures; but such union is effected so slowly that, for all purposes for which the fuel is required, it may be considered as practically non-existent.

Spontaneous Ignition.—The chemical combination of combustibles with oxygen at low temperatures must not be ignored. Coals exposed to the action of the open air, sun, and rain become deteriorated by slow combustion. If totally immersed, there is some reason to believe that the calorific value may become greater; for some New Zealand coal gives out a greater quantity of heat than the best Welsh, notwithstanding a more than moderate amount of wetness. For slow combustion to become active, it is necessary that the air should have slight access to the fuel, and be almost stagnant. A small quantity of heat is then generated by slow combustion, and the consequent rise of temperature fosters the same process between other particles; and the oxygen contained in any dampness present assists the process. These combinations proceed with increasing rapidity as the temperature increases, until some disturbance takes place, admitting more air, when the carbon gases evolved burst into flame, in some cases explosively.

Furnace Combustion.—It is only at a high temperature that a common fire burns, and if from any cause the temperature of the surrounding substances be lowered, the fire will cease to burn briskly. When coal is heated gases are given off, represented by the chemical symbols—

CH_4 = light carburetted hydrogen or marsh gas, fire-damp.
C_2H_4 = heavy „ „ olefiant gas.
C_2H_2 = acetylene
C_6H_6 = benzine
$C_{10}H_8$ = naphthaline
} and others of similar composition.

The first two are the most important; and, as far as the furnace of a boiler is concerned, the others may generally be ignored.

In a boiler furnace, under ordinary conditions, there are various distinct stages of combustion.

1. If Welsh coal be heated (bituminous coal is used in the ordinary production of gas for lighting purposes), over 10,000 cubic feet of hydrocarbon gases are distilled off from each ton of coal so treated; this leaves a solid residue, which is practically all carbon. Thus, when coal is thrown on the fire in the furnace heat is first

used to distil off these gases. At a sufficiently high temperature hydrogen gas is also given off.

2. If the temperature be sufficiently great these gases first separate into carbon and hydrogen, then chemically combine at a temperature of about 1400° Fahr. with the incoming oxygen of the air, and form oxides of carbon and hydrogen; combustion ensues, producing **heat.** All the air necessary for this stage, if supplied at a sufficiently high temperature, might be injected above the fire—that is, through the furnace door, or thereabouts.

3. The air supplied below the fire is at first warmed by the fire, then the oxygen in it combines with particles of solid carbon on its way through the fire, generally forming carbon monoxide and dioxide (CO and CO_2); and if the temperature be high, together with a sufficiency of oxygen, the CO is converted into CO_2 before it leaves the furnace.

The **funnel gases** are generally carbon dioxide, or CO_2; steam gas, or H_2O; and nitrogen, or N, which forms about 77 per cent by weight of the air supplied for combustion.

Rate of Combustion.—The weight of fuel burnt per square foot of grate surface per hour is called the rate of combustion.

In the Merchant Service, when working economically with funnel draught only, the rate is from 15 to 20 lb. of coal. It is now more usual to adopt some form of forced draught—Howden's closed ashpits is most frequent—or of induced draught, when the rate is increased to 24 to 40 lb.; but at the same time the length of grate is frequently decreased, so that the power per unit of boiler is not increased to quite the same extent. The improved systems of forcing have during the last few years, however, increased the power obtained from similar boilers by 40 to 60 per cent, and in one or two instances by as much as 90 per cent.

Lighting Fires.—When lighting fires in a boiler of the cylindrical type, with more than one furnace, the lower fire or fires should be lighted up before the others; generally two or three hours are allowed to warm the boiler gradually by promoting sufficient convection current for the purpose. The fires inside the several doors of a water-tube boiler should all be lighted at the same time.

The furnace is *primed* by throwing on coal to an even thickness of about 3 or 4 inches. Just inside the furnace door wood and coal are topped up ready for lighting, and should be so arranged

that all air entering the furnace door must also pass through the topping. A piece of oily waste is generally used for lighting; the furnace door is left partly open and the ashpit door closed until the topping is well alight and ready for spreading and pushing back, which is done with a rake. After spreading the fire the furnace door is closed and the ashpit door partly opened, and the draught regulated by the amount of opening. Fresh coal is added as necessary, and the fire gradually levelled up to a thickness of about 6 inches.

Funnel Draught.—Before lighting fires, a draught or current is created by the air in the boiler room being somewhat hotter, and therefore lighter per gallon, than the air at the mouth of the funnel. The lighting of the fires creates a still stronger current, because the temperature inside the funnel increases. The current is also increased by additional height of funnel. It is wasteful to allow the funnel temperature to become too high, because heat is then passed away into the atmosphere and no useful work obtained from it. Thus when this temperature is moderate, between 600° and 700° F., the draught is practically dependent on the height of the funnel above the firegrate, and if no artificial means are employed to assist the draught it is termed *Natural draught.*

In destroyers and other small vessels only a very little height of funnel can be fitted, and some system for increasing the draught —called *Forced draught,* but actually assisted draught—is used.

Forced Draught.—The primary object of forced draught is to burn more coal or fuel per square foot of grate surface, and thus obtain a greater power from a given weight, and also on a given floor area, without increasing the funnel temperature and consequent loss of heat.

The principal methods in ordinary practice are:—

1. Closed stokeholds, when the only outlet for the air under compression is the firegrate. (Plenum.)

2. Closed ashpits, when the stokeholds are open to the atmospheric pressure, but the furnaces are supplied with air, compressed, and generally heated previous to supply. (Plenum.)

3. Suction draught, by drawing up the products of combustion from the furnaces through the usual passages. (Vacuum.)

4. Steam or exhaust blast, inducing and accelerating the draught in the funnel. (Vacuum.)

5. Jets of compressed air blown into the base of the funnel and

inducing a draught, now practically abandoned as too cumbersome. (Vacuum.)

6. Jets of liquid fuel blown over the incandescent coal. This method increases the rate of combustion, but the air is not necessarily supplied artificially.

1. **Closed Stokeholds.**—This system has been frequently adopted for increasing the rate of combustion in warships. It is a ready method of increasing the power obtainable, but it is the *least economical* of all means of promoting artificial draught. For warships, where the full power, however, is seldom required, it answers the purpose for which it is fitted; but in the mercantile marine, where the primary object is to obtain an economy of fuel, it is very properly excluded. The extra weight is less than that required with either suction, closed-ashpit, or the compressed-air systems. Water-tight compartments already exist in warships, and the only additions necessary are a few thin screen doors of sufficient strength to resist the very light air-pressure of the excess, or plenum, over that outside the screens. Air locks (as shown in Fig. 105, for twelve Yarrow boilers fitted in *Triumph*) with double doors are usually fitted for entry to and exit from the stokehold, so that no sudden decrease of pressure should produce a back draught from the furnaces. An objectionable feature of the closed stokehold is that the men on watch are shut in, and outside supervision is very difficult.

A necessary adjunct of the closed stokehold is that the air-supply must be cold, although slight heating is effected in some boiler installations of the Thornycroft and Yarrow types. This does not prevent the admission of considerable quantities of cold air when the fires are cleaned. The admission of cold air has been disastrous to cylindrical boilers when forcing above a rate of combustion of about 22 lb. per square foot of grate, especially in boilers of the common combustion-chamber type, and also in boilers of the water-tube type, with straight tubes, nearly horizontally arranged, with little allowance for independent expansion of the tubes. Such failures are caused by lack of free circulation in the boiler and lack of proper allowance for difference of expansion in the various parts.

2. **Closed Ashpits.**—Apart from the supply of hot air to the furnaces for combustion, this system has some advantage over others, and in combination with such heating it is the most economical. Howden's system of closed ashpits and hot air-supply

has been extensively tried and proved. It is fitted in a large number of ocean-going steamers. In almost every case there has

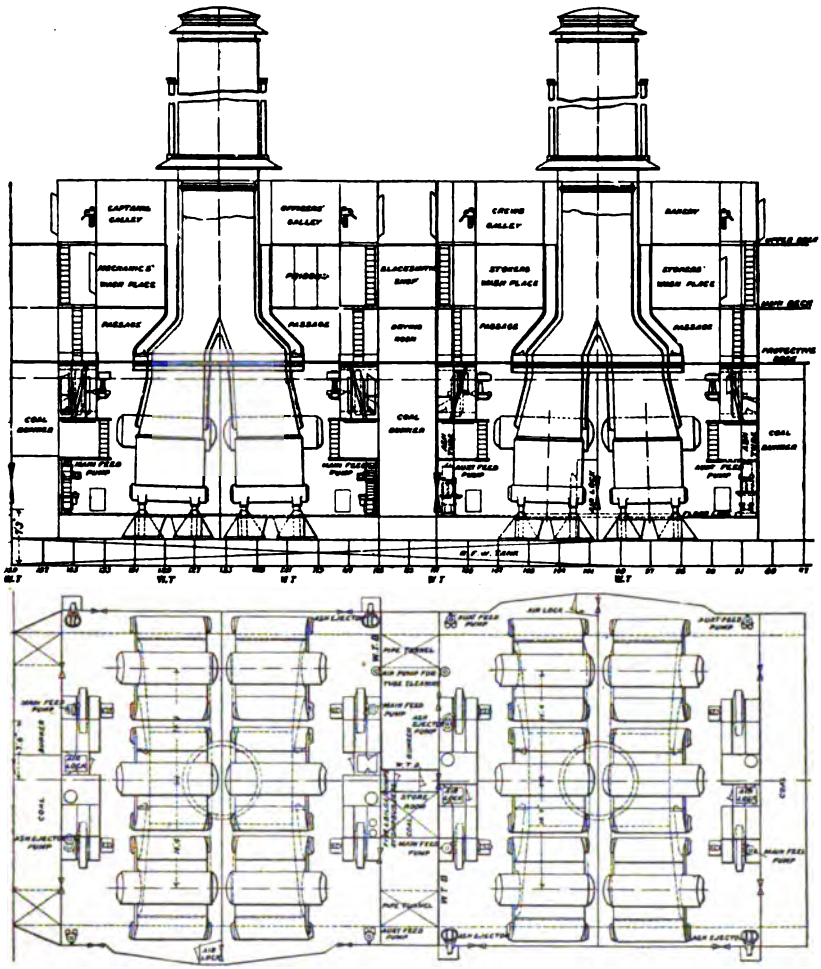


FIG. 105.—Closed Stokeholds (*Triumph*).

been a gain in economy of fuel, and in many a saving in space has also been effected.

The hot air-supply system consists essentially in a number of very thin tubes fitted vertically in the smoke box just above the exit of the gases from the boiler heating tubes, as shown in Fig. 106. The gases pass through these tubes, internally, on their way

to the funnel, and give up a large amount of their heat to them, which in turn pass it on to the air-supply from their external surfaces. The air-supply to the boiler-room comes down the usual ventilators (either with or without cowls on top), and is forced by a fan, through a trunk delivery, among the tubes, externally; thence downward through passages provided around the casing of the smoke

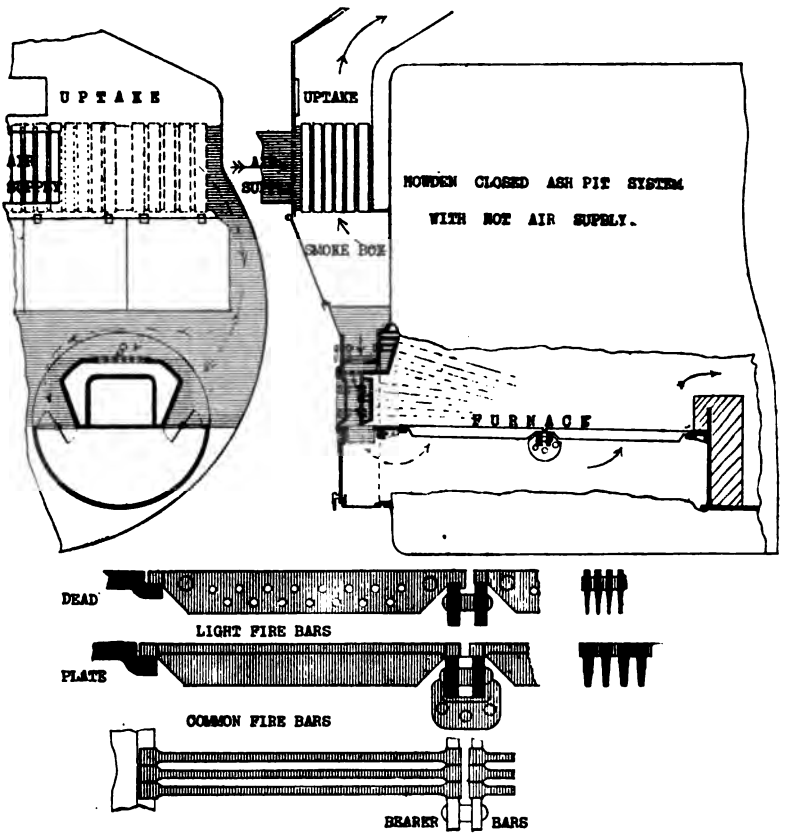
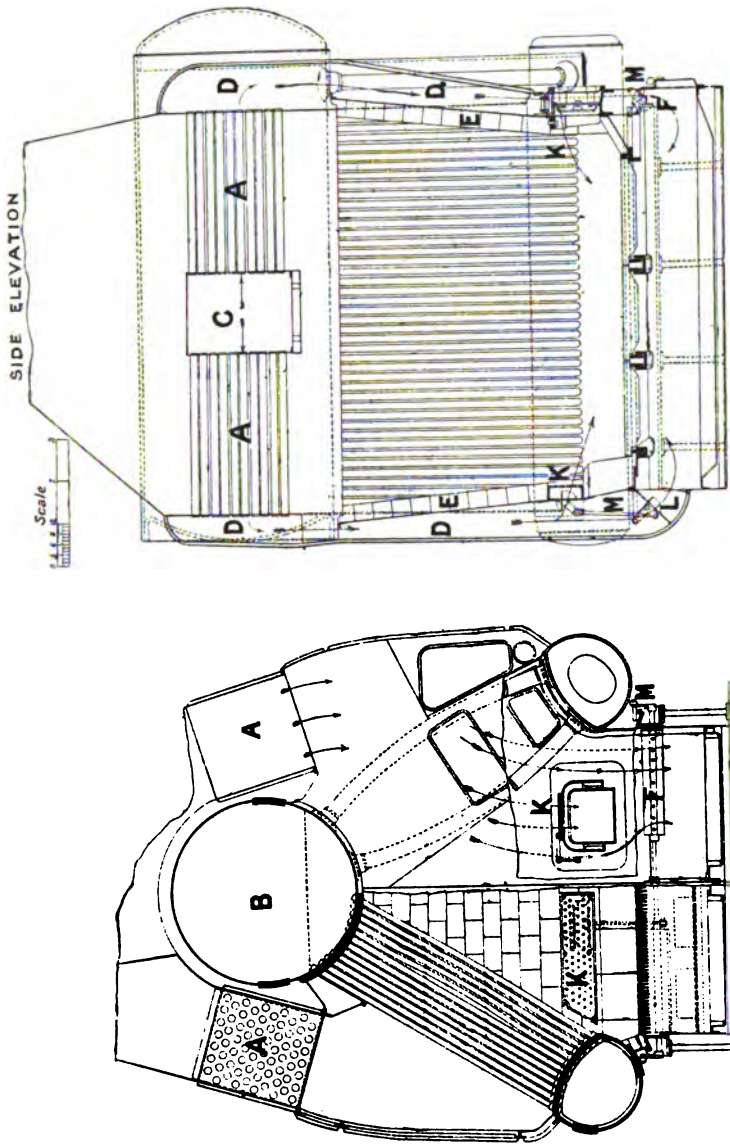


FIG. 106.—Closed Ashpits.

box and in front of the boiler, and through holes in the furnace front both above and below the firegrate, in the proportion of about 1 to 3. By a simple system of levers it can be arranged that, when either the furnace or the ashpit door is opened, the draught can be automatically and independently shut off—thus a possibility of accident is easily prevented; but this fitting is usually unnecessary.

If the boilers are suited to high rates of combustion, the system can be fitted conveniently to water-tube boilers.



Figs. 107 and 108.—Closed Ashpits and Hot-air Draught.

H.M.S. *Antrim* is fitted with Howden's system of closed ashpits and hot air-supply in conjunction with Yarrow boilers, one of which

is shown in Fig. 107. There are four nests, *A*, of air-heating tubes fitted in the uptakes, two on each side of the steam drum, *B*. Air is drawn through the boiler-room and ventilators, and pumped by centrifugal fans through trunks into the space *C*; the air then passes through the tubes, *A*, into the front and back end casings, *D*, of the boiler, between the brickwork casing, *E*, and the outer casing, from which it supplies the fire above the grate (at *KK*), and below (at *L* and *F*) through air-doors and slides. By a simple system of levers, *M*, connected with the hinges of the fire-doors, the air-supply is automatically shut off while a fire-door is open. The ashpit doors are always closed except when drawing ashes, etc., and the pressure in the ashpit may be (equal to 2 or 3 inches of water in some cases) in excess of the pressure (atmospheric) in the stoke-hold. The temperature of the air-supply to the furnace is from 180° at ordinary powers up to 300° or more at full power, and naturally a high economy of fuel should result because the heat supplied to the air is taken from the funnel gases. The air-heating surface is about thirteen times the grate area, and the tubes $2\frac{1}{2}''$ diameter.

Great care should be taken in fitting up the lead of pipes from the ventilating trunks to the furnaces, in order that the loss of head, due to friction in the pipes, bends, etc., shall be as small as possible; otherwise the increased power required to drive the fan increases the expenditure of fuel unnecessarily, and the fan itself becomes cumbersome, and very liable to break down. The lead should be fair, with as little abrupt alteration of direction of current as possible. The trunk ventilators, with right-angle branches, originally fitted in so many of His Majesty's ships, are a good example of what to avoid.

The **supply of hot air** to the furnace tends to reduce the quantity practically necessary for combustion; probably 18 to 20 lb. of air per 1 lb. of coal are sufficient. This reduces the velocity of the furnace gases on their way to the funnel, and, allowing them more time in which to impart their heat to the boiler, consequently reduces the funnel loss. It may be taken that, approximately, any reduction in the funnel temperature which is employed in heating the furnace supply is a double gain, and thus the efficiency of combustion is greatly improved: the furnace temperature is higher, greater heat can be imparted to the water, and combustion is more complete. The reduction of funnel temperature and the casing in

of the smoke boxes by the screen plates for conducting the downward flow of the warmed air, the temperature of which is considerably lower than that of the smoke box, conduce to a much cooler ship and stokehold, which is much appreciated in hot climates.

3. **Suction Draught.**—The great advantage of this system—in fact, any system which does not involve entirely shutting off the stokehold—is that the stokeholds are always open, comparatively clean, cool, and comfortable, and the ventilation of these parts is better. In the system tried in two battleships there is stated to be some economy of fuel; but it is very small, as no method of heating the air required for combustion is included in the installations, and it is in this way that suction draught is productive of great economy as compared with the closed stokehold system. The situation of the suction fans in the base of the funnel increases the wear and tear, and the volume of the hot gases passing through the fan is nearly double that in other systems with cold air-supply. This might be objectionable; but if the air is heated to nearly the same temperature as that of the escaping gases, then it appears that the volume of gases is practically the same in any system, and that the above statement requires some modification. Possibly only about 10 per cent more power may be required to drive the larger fans for the suction system as opposed to, say, Howden's hot-air and closed ashpit system. Notwithstanding the mechanical difficulties attending the efficient working of suction fans, there is some gain more than can be accounted for, except that in nature a pull is generally more effective than a push.

In the Ellis system, which is shown in Fig. 109, the fan is placed in the base of the funnel, and there is an alternative passage around the damper when the fan is not required. The air is drawn into the furnace both above and below the firegrate, as shown by the arrow-heads, and the gases follow the usual course through the combustion chamber and tubes to the smoke box. From here they are drawn through the passage-ways, externally, to a number of very thin tubes arranged above the boiler (as shown in the figure), and thence to the fan, which delivers them into the funnel.

The air-supply is sucked into these heating tubes, internally, from the stokehold, and also from any other opening which is in connection with the intake, thence by way of passages around the smoke boxes to the furnace. Unless the increase in weight due to the extra fittings is compensated by a corresponding increase in

the power obtained, this might be an objectionable feature; but there is not necessarily any increase in a well-arranged system.

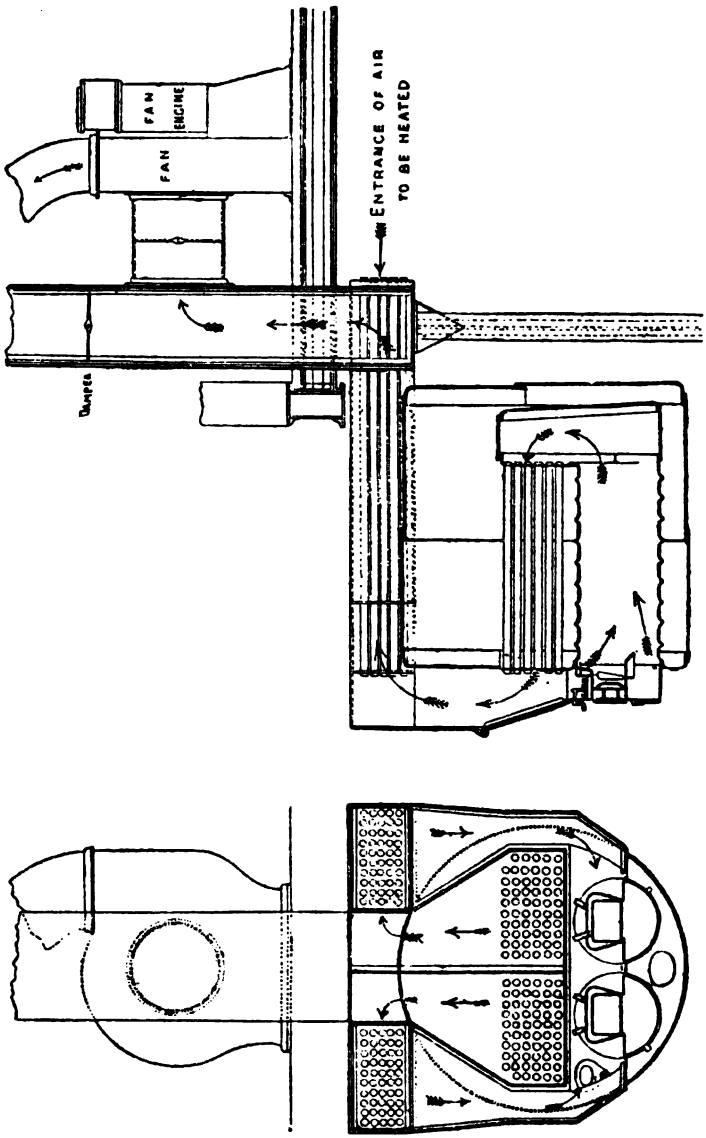


FIG. 109.—Suction Draught.

The substitution of helical fans, similar to a screw propeller, working on a vertical shaft, appears to be far superior to the

centrifugal type now in use. The position of these helical fans could be kept lower, and the engine could also be fitted in a more suitable position for efficient working, while the direction of movement of the gases would not then be subject to the abrupt change, as shown in the figure. When the helical fan is not required, it would not offer serious resistance to the passage of the gases through it.

4. **Steam** *or* **Exhaust Blast.** — The steam blast is a ready method of inducing a draught and thus increasing the rate of combustion. Its use has been generally abandoned for marine purposes because it is so wasteful both in fuel and water.

The exhaust blast, as used in locomotives, is probably the most prolific method of increasing the rate of combustion. The highest known amount, viz. from 200 to 224 lb. per square foot of grate, is seldom attained, but 120 lb. is common. This rate does not produce any immediate defect in the boiler, but it no doubt shortens its life. The exemption from immediate defects is entirely due to the mode and manner of working and forcing. The draught produced by the exhaust is exactly proportional to the power taken from the boiler. The damage which generally follows from the admission of cold air is also reduced to a minimum, because the stoking is almost entirely done when the engine is stopped, or at low speeds and powers. No fire-cleaning or other operation which necessitates the prolonged opening of the fire-door is necessary while the engine is running at full power and with the blast full open, for this last condition is generally required only for short periods; this practice is entirely impossible in sea-going ships. The water required to make up that lost up the funnel is very great, probably over 70 per cent of the total water evaporated, because only a small amount is deposited in the exhaust tank. The fires must be stoked to maintain a continuous supply of steam for full power, which in marine practice entails cleaning them at frequent intervals, when cold air must also be admitted, resulting in serious damage to the boiler.

Air-Supply.—The supply of air below the fire must be properly regulated. The fire bars must be spaced a proper distance apart, according to the type of fuel used. For the best Welsh coals, which are quick-burning, the area of opening between the bars for moderate rates of combustion is generally about 50 to 55 per cent of the total fire area. With North-country and N.S.W. coals,

which are relatively slow-burning, the proportion of spacing should be decreased to about 35 or 45 per cent. For forced draught with Welsh coal the proportion of opening is increased to 60 per cent.

To allow the air sufficient passage through the fire, the average thickness of the fire should never exceed 6 inches for quick-burning coals, and about 8 inches for slow-burning coals.

If the air-supply to the furnace is heated before it reaches the fire, it tends to reduce the formation of smoke, because the consequent increase of furnace temperature produces more complete combustion.

Gas Mixers.—With Belleville boilers the tubes form the upper part of the furnace, and consequently there is a great tendency for the gases to become cooled below the temperature necessary to promote further combustion, owing to the absence of sufficient room for combustion. Air jets are fitted near the furnace doors as shown in Fig. 64, to play almost at right angles (*i.e.* horizontally) to the upcast gases from the fires. These air jets assist in the completion of combustion; and, if the pressure be properly adjusted, will prevent smoking, and at the same time promote economy. They baffle the draught, and allow time for the gases to be properly burnt. Although they assist in the production of a greater total evaporation, these jets do not constitute forced draught in the proper sense of the term, but form merely a part of the boiler plant for a special purpose. About one-fourth of the air required can be admitted above the grate, as mentioned elsewhere. If the pressure in the air jets is too little, the air does not reach the back end of the fire, and incomplete combustion results with smoke or flaming. If the pressure is too great, too much air may be supplied and the fire temperature reduced, and smoke is produced.

The Niclausse boiler is now usually fitted with gas mixers.

The pressure in the jets is about the following for good combustion:—

When burning	10	lb. of coal	per	sq. ft.	of grate,	6 lb. per sq. in.
„	15	„	„	„	10	„
„	20	„	„	„	13	„
„	24	„	„	„	15	„
„	30	„	„	„	17	„

The fire in these boilers should be kept about 5 inches thick for high rates of combustion

Stoking.—Economy of fuel can be obtained by careful and regular stoking, and the coal should be handled with a shovel, care being taken to have the coal small enough to part with its gases easily, and at the same time not small enough to fall through into the ashpit. The coal should be thrown on evenly, about four or five shovelfuls at a time, and the average thickness of the fire should not be greater than 6 inches. With a fire of 5 inches thickness the supply of air should be sufficient to check smoking, and at the same time give sufficient air for complete combustion of the fuel.

Under forced draught a *thin* fire is also necessary for good combustion. The air pressure, or *plenum*, can be considerably reduced by efficient stoking, and for this reason the amount of air pressure is a very unreliable measure of the rate of combustion, unless the fires are well stoked and are kept a proper thickness. Good stoking alone will increase the indicated horse-power or evaporation of a boiler by at least 10 per cent, and instances have been observed where the increase has been over 25 per cent.

The fires should be levelled with a rake about every half-hour, and *holes in the fire* should be guarded against, because a large volume of cold air may be admitted in this way. If the gases go up the funnel as carbonic oxide, only 4400 thermal units are given out; but if it can be properly burnt by a sufficient supply of oxygen, 14,500 units can be imparted to the boiler for each 1 lb. of coal; therefore, within ordinary limits, it is better to have too much rather than too little air supplied. Because the coal has disappeared from the grate it does not necessarily mean that it has been usefully burnt; both carbonic oxide and carbon dioxide are invisible, and thick fires, through which a sufficient supply of air cannot penetrate to the part above the fire for combustion, should be carefully avoided.

The fire bars should be cleared by a pricker from below the grate, not from above, because this entails opening the fire-door and admitting cold air to the furnace, with an attendant lowering of the furnace temperature. The furnace doors should be kept closed, and opened for as short a time as possible when firing. When cleaning the fire the ashpit damper should be kept closed; in short, the furnace door and the ashpit damper should never be open at the same time. A routine of firing should be laid down suitable to the boilers, and always carried out, so that the green coal does not entirely cover the grate.

An accumulation of ashes is a thing to be guarded against, and they should be drawn out at least once in each four-hour watch; they should then be sifted or riddled, and the small coal thrown back on the fire. Ashes settle in or among the tubes, and these must be removed at regular intervals; and, as opportunity offers, the heating surfaces should be cleaned. Soot is a very bad conductor of heat, and consequently any slight scale on the heating surface considerably reduces the evaporative power. For cleaning the interiors of the tubes in cylindrical boilers, forked scrapers or button-ended scrapers are used, and steel-wire brushes are at all times most useful. The removal of *retarders*, which now form part of cylindrical boiler plant, also materially assists in keeping the tube heating surfaces clean. After removal, these should be cleaned with a wire scrubber before they are replaced.

For boilers of the Belleville, Niclausse, Dürr, and Babcock types, half-moon scrapers with long handles can be used for cleaning the exteriors of the tubes; and wire brushes are also used, whenever practicable, for both the interior and exterior surfaces of the tubes of all water-tube boilers.

When *banking fires*, they should be banked in front if banked at all; generally this is unnecessary. The fires should be allowed to burn down as low as possible, and a little fire kept alight at the front end, in preparation for pushing back when required. If the fires are dirty, the clinker should be taken out after breaking it up by a pricker applied below the grate; but the removal should not be carried out until the fire is burnt down very low, to reduce the possible damage from the admission of cold air. Care should be taken that the fire is kept well back from the dead plate at the furnace door. The intelligent carrying out of these details tends to the reduction of strains on the boiler and freedom from leakage, especially at the furnace and tube-end joints.

Smoke.—Smoke is the *visible* result of incomplete combustion of the tarry constituents of the fuel, and in practice is caused by the temperature of combustion being below that of the ignition-point of these tarry constituents. Of the three gaseous products—steam, carbon dioxide, and carbon monoxide—of combustion, steam alone plays an important part in the formation of smoke, whilst the other less important constituents are tar vapour, minute particles of unburnt carbon, and ash, drawn upwards by the funnel draught. The amount of carbon in smoke is far less than is generally

supposed, even when given off by bituminous coal under the imperfect conditions of combustion existing in an ordinary fire-place. (See *Cantor Lectures*, 1906, Professor V. B. Lewes.)

Smoke consists of vesicles, globules, or bubbles of nitrogen, carbon dioxide, carbon monoxide, traces of oxygen and such hydrocarbons as methane, enclosed in skins of condensed vapour, and liquids from the burning substances which give rise to them. If collected, and the tiny vessels (which are very light, and float easily in the atmosphere) are burst by friction, a liquid "tar" is obtained and the gases escape. In the production of ordinary lighting gas the tarry liquids are removed by distillation at a comparatively high temperature, and the coke remaining, and consisting of almost pure carbon, burns smokelessly, but only ignites and continues to burn at a high temperature. But if the temperature of distillation be comparatively low and only just sufficient to expel the tarry constituents, principally in the liquid state, an intermediate form of coal and coke, now known as "Coalite," remains, which ignites at a comparatively low temperature and is stated to produce no smoke.

Soot is produced when a strong draught carries up the gaseous bubbles and particles of carbon dust. Friction against the sides of the flue and chimney passages bursts the bubbles, and the liquids of which the skins are formed adhere to the surfaces in conjunction with carbon dust. The liquids are generally vaporised or burnt away, leaving almost pure carbon dust in the form of soot in various quiet places formed by the flue passages.

In practice the *prevention of smoke* formation from coal and oil fuel is attainable only by careful stoking, ensuring good combustion at a temperature high enough to completely burn the tarry matters contained in the fuel. Boilers which are most likely to produce smoke are those in which the space for combustion is unduly contracted, and in which the furnace height is small, and where, as in the Belleville, Dürr, Niclausse, Babcock and Wilcox, and other boilers generally of the horizontal water-tube type, the cold surfaces of the generating tubes are in close proximity to the fire. The gas mixers, as to Belleville boilers, when properly used tend to reduce smoke formation; but it is quite possible to produce excessive smoke by supplying too much air, and thus reducing the furnace temperature by this means below that necessary for proper combustion. Generally it may be considered that the gas mixers near the furnace doors of Belleville boilers tend to reduce smoke formation, and those

fitted to supply air to the combustion space between the top of the generator and bottom of the economiser of Belleville boilers tend to reduce flaming. The upper gas mixers with ordinary care are not necessary to good combustion, and if used might be detrimental.

Flaming.—An unnecessarily thick fire does not allow sufficient air to pass through it to complete the combustion of the gases given off by the fuel. Carbon monoxide, CO, is thus produced, and this *invisible* gas at a high temperature bursts into flame when it comes into contact with more air, including oxygen, at the mouth of the funnel; the CO burns into CO_2, carbon dioxide, and produces the flames seen. Flaming can be almost instantly stopped by slightly opening the furnace door, thus increasing the air-supply above the fire; and an opportunity should at once be made to level off the fires, to clean them if necessary, and to reduce them to an even thickness. Flaming is most likely to occur with water-tube boilers without economisers, unless the gas mixers are properly used, but is most noticeable in small vessels with short funnels. It is seldom noticed with cylindrical boilers because the gases have a longer and more circuitous route to travel, which tends to reduce the temperature at the funnel mouth below that at which the carbon monoxide will take fire. Accidental flaming may occur for an instant when under a strong forced-draught; but, as pointed out above, it is always preventible, it can be instantly checked when known, and it is entirely dependent on the stoker.

Periodical Cleaning of Fires.—In practice the fire requires cleaning after about 2 cwts. of coal have been burnt on each square foot of grate surface; the time and the amount depend on the rate of combustion, and also the class of coal used. The heating surfaces, on the side next the fire and furnace gases, require cleaning after about 6 cwts. have been burnt per square foot of grate. The necessity for cleaning the fire limits the time for which the full power of all the boilers can be maintained. Thus, a destroyer burning 60 lb. or over per square foot of grate is limited for full power to about 3 or 4 hours; a battleship or cruiser burning 30 lb. is limited to about 8 hours—or, burning 15 lb., to 16 hours—without cleaning fires. But, of course, by a sufficient reserve of boiler power the full *engine* power can be obtained for any period. Generally the boiler power of naval vessels is sufficient to maintain 75 to 80 per cent of the full engine power, and the reduction of sea-going speed is generally not greater than about 5 per cent.

CHAPTER XI

EFFICIENCY AND THEORETICAL CONSIDERATIONS

Calorific Value.—This is the quantity of heat which is given out by 1 lb. of the fuel when combustion is complete; it is measured in British thermal units.

The actual calorific value can only be found by experiment; a calorimeter is used for this purpose, but approximate calculation is usually based on the following assumptions.

The quantity of heat generated by the combustion of 1 lb. of carbon is found by experiment to be about 4400 b.t. units if converted into CO; or about 14,500 units if converted into carbon dioxide—and, if the air-supply is sufficient, it may be concluded that all the carbon is so converted.

The quantity of heat generated by the combustion of 1 lb. of hydrogen is found by experiment to be about 62,032 b.t. units; but this assumes that the steam produced by the combustion is subsequently condensed, and the latent heat transferred to the boiler. Such is not the case in practice, because the temperature of the escaping funnel gases is always greater than 212°, the boiling-point of water at atmospheric pressure; and therefore the latent heat and quantity of superheat must be subtracted from the heat of combustion. One pound of hydrogen combines with 8 lb. of oxygen to form 9 lb. of steam or water, and each 1 lb. of H_2O carries away at least 966 b.t. units as uncondensed steam. Therefore the heat generated by the complete combustion of 1 lb. of hydrogen in an actual boiler approximates to $62{,}032 - 9 \times 966$, or 53,338 b.t. units.

The following calculations neglect the heat absorbed by the dissociation of carbon and hydrogen; and the fractions of carbon, hydrogen, and oxygen contained in the fuel are represented by their chemical symbols, C, H, and O.

The actual amount of hydrogen available for combustion is $\left(H-\frac{O}{8}\right)$, because $\frac{O}{8}$ lb. of the hydrogen are already combined. Thus the calorific value is, approximately, in b.t. units per 1 lb.—

$$= 14{,}500C + 62{,}032\left(H-\frac{O}{8}\right)$$
$$= 14{,}500\left\{C + 4{\cdot}28\left(H-\frac{O}{8}\right)\right\}$$

if the steam is condensed before passing up the funnel. Or, if the steam passes away without condensing, which is most probable—

$$\text{Calorific value} = 14{,}500C + 53{,}338\left(H-\frac{O}{8}\right)$$
$$= 14{,}500\left\{C + 3{\cdot}68\left(H-\frac{O}{8}\right)\right\}.$$

Evaporative Unit.—For purposes of comparison of the calorific value of various fuels, a derived unit is employed. It is the weight of water which can be theoretically evaporated by 1 lb. of the fuel, from a temperature of 212°, at the same temperature, and under ordinary atmospheric pressure.

The number of thermal units required to evaporate 1 lb. of water, from and at 212°, is 966. By adopting this temperature the sensible heat is omitted, and the evaporative unit is the calorific value divided by 966; that is—

$$= \tfrac{14500}{966}\left\{C + 3{\cdot}68\left(H-\frac{O}{8}\right)\right\}$$
$$= 15\left\{C + 3{\cdot}68\left(H-\frac{O}{8}\right)\right\} \text{ in b.t. units per 1 lb. of fuel.}$$

Applying this to the proportions for average Welsh coal, the evaporative unit—

$$= 15\left\{84 + 3{\cdot}68\left(5-\tfrac{1}{2}\right)\right\} \div 100$$
$$= \tfrac{15}{100}(84 + 4{\cdot}5 \times 3{\cdot}68)$$
$$= 15 \text{ lb. about, for each lb. of Welsh coal.}$$

In practice this is seldom obtained; a fair average result is only about 9 lb., but under good conditions as much as 12 lb. of water

are evaporated per 1 lb. of good Welsh coal. For Admiralty tests *on shore*, the amounts required are—

11 lb., when burning not less than 30 lb. per sq. ft. of grate.
11½ " " " 24 " " "
12 " " " 18 " " "

Air-Supply *by weight.*—Oxygen is found mechanically mixed with nitrogen in atmospheric air in the proportion by weight of about 23 to 77, so that for the supply of 1 lb. of oxygen 4·35 lb. of air are required. For complete combustion 1 lb. of carbon requires 2·66 lb. of oxygen, or—

$$4{\cdot}35 \times 2{\cdot}66 \text{ lb.} = 11{\cdot}6 \text{ lb. of air.}$$

For complete combustion 1 lb. of hydrogen requires 8 lb. of oxygen, or—

$$4{\cdot}35 \times 8 \text{ lb.} = 34{\cdot}8 \text{ lb. of air.}$$

But part of this hydrogen is already contained in the fuel itself, and only $\left(H - \frac{O}{8}\right)$ lb. require air supplied, so that—

Total weight of air required for complete combustion of 1 lb. of fuel $= 11{\cdot}6C + 34{\cdot}8\left(H - \frac{O}{8}\right)$

$= 11{\cdot}6\left\{C + 3\left(H - \frac{O}{8}\right)\right\}$; or

= 11·3 lb. for about 1 lb. of Welsh coal.

In practice this amount is insufficient, and from 15 to 24 lb. must be supplied respectively for a very efficient artificial draught, and for natural draught, due to height of funnel only.

The *volume* occupied by 1 lb. weight of air at ordinary atmospheric pressure and at a temperature of 62° Fahr. is about 13·14 cubic feet, and therefore about 177·1 cubic feet of air must be supplied for each 1 lb. of coal burnt under artificial draught, and 315·4 cubic feet for funnel draught. And for *each ton* of coal burnt under the above conditions, 396,704 and 706,406 cubic feet are required respectively.

These volumes of air must be considered when calculating the passage areas of ventilators, ashpit doors, etc., and if the available

areas are too small, the velocity of the air passing through them must be artificially increased.

Funnel Draught.—A fire, whether of coal or other fuel, must be supplied continuously with oxygen for the continuous combustion or burning of the fuel. This oxygen is derived from atmospheric air, as already stated, and it is now proposed to investigate the natural cause which continues the supply of air. If there are two bodies of fluid in separate chambers but of differing densities, and originally at the same level, the heavier fluid, when a connection is opened between the lower ends of the two chambers, gradually displaces the lighter by pushing it upward. The rate of flow between the two chambers (neglecting frictional resistances) is governed by the difference of pressure (or head, or weight per cubic foot or unit of volume) on each unit of area. When a common fire is burning, the *natural draught*, or rate of flow up the chimney, is caused by the convection current produced by a difference of pressure forcing through the fire a continuous supply of fresh air to the fuel.

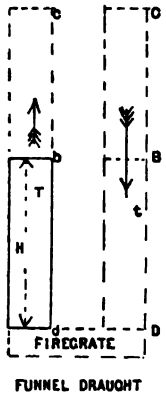


FIG. 110.—Funnel Draught.

In Fig. 110, let DC (= *dc*) represent the height of the atmosphere above the firegrate.

„ DB (= *db*) represent the height of funnel above the firegrate.

And let A = area of each of the columns DC and *dc*
= area of funnel.

Then, weight of air in column DC = weight in columns (BC + DB),
and „ „ „ *dc* = „ „ (*bc* and *db*),
in which the weight in column BC = weight in column *bc*.

Difference in weight = weight in column DB − weight in column *db*.

If there be no wind or cross current at the top of the chimney or funnel, it is possible that the weight of the column *bc* may be less than that of BC, because of its higher temperature; but for all practical purposes this difference is negligible. It is concluded, therefore, that the velocity, or rate of flow, inside a chimney or funnel is due to the head of pressure created by the difference in

weight of two columns of ascending and descending air or gases of the same height and area as the ascending column. For all practical purposes, *but only* when both are at the *same temperature* and pressure, the weight of the products of combustion per cubic foot is equal to that of a cubic foot of atmospheric air.

Let the height of funnel above the firegrate be H in feet.
area of funnel opening „ A in sq. ft.
absolute temperature of funnel gases „ T_0.
„ „ air-supply „ t_0.
velocity of gases inside funnel „ V in feet per second.
weight of gases passing through funnel „ W in lb. per second.
„ 1 cubic foot of funnel gases „ u in lb.
„ „ „ air-supply „ w in lb.
pressure of atmosphere at top of funnel „ q in lb. per 1 sq. ft.

Then weight of gases inside funnel $= A . H . u$.

pressure outside funnel, at fire-grate level $= q + A . H . w$.

pressure inside funnel $= q + A . H . u$.

Difference of pressure outside and inside when gases are not in motion

$$= A . H (w - u);$$

but in practice the gases are in motion upward, and the force producing motion = M . V. =

$$\text{Mass} \times \text{velocity} = \frac{W}{g} \times V = A . H . (w - u) \qquad (1).$$

In which $W = V . A . u$.

Therefore $$\frac{V^2 . A . u}{g} = A . H . (w - u),$$

and $$V^2 = \frac{(w - u)}{u} \cdot g . H \qquad . \qquad . \qquad (2).$$

Now, 1 cubic foot of air at 493° abs. F. weighs 0·08073 lb.

Therefore $$w = \frac{493 \times 0{\cdot}08073}{t_0} \text{ (from Charles' Law, P. V. } = c . T_0),$$

and $$u = \frac{493 \times 0{\cdot}08073}{T_0}.$$

From which $$\frac{w - u}{u} = \frac{T_0 - t_0}{t_0} \qquad . \qquad . \qquad . \qquad (3).$$

Substituting in (2),

$$V = \sqrt{\frac{T_0 - t_0}{t_0} \cdot g \cdot H} \quad . \quad . \quad . \quad (4).$$

EXAMPLE.—When $T_0 = (585 + 461) = 1046°$ abs. F.,
and $t_0 = 62 + 461 = 523°$ abs. F.,
then $T_0 = 2 \cdot t_0$
and $V = \sqrt{g \cdot H} \quad . \quad . \quad . \quad . \quad (5).$

In many recent battleships H = 82 feet, and, therefore, with the above temperatures, V = 51 feet per second; but it should be clearly understood that this is only a relative value, and does not take into account the frictional and other resistances which may reduce the velocity by more than one-half.

From certain calculations, into which it is not necessary to enter, it can be shown that the *weight of gases* which can be discharged per second through the funnel is a maximum, when $\frac{T_0}{t_0} = \frac{25}{12}$, approximately.

Thus, if $t = 100°$ F., which is a common temperature, the maximum weight of gases is discharged when $T = 707°$ F. $(= 1168°$ abs. F.). Notwithstanding this fact, it is possible in practice to increase the rate of combustion and quantity of coal burnt by increasing the funnel temperature, because with the increased velocity so produced a less weight of air may be required per 1 lb. of coal. This method, however, increases the funnel loss, and in addition decreases the boiler efficiency from incomplete combustion, increased radiation, and inferior quality of steam.

Damaged Funnel.—If the funnel is damaged and penetrated, its virtual height, H, is decreased, and the draught, otherwise created by it, is partially or wholly destroyed above the height of the lowest hole. Small holes have, of course, a less effect than large ones, and the proportion of the draught destroyed depends on their position and on their area relative to that of the funnel passage-way. In the absence of sufficient forcing arrangements, and after the funnels are damaged, the rate of evaporation, on which the engine power depends, can only be maintained by raising the funnel temperature. In one instance this is stated to have risen to 1800° F., and the coal consumption to have been doubled for the same power.

Mechanical Draught.—For naval purposes an increased rate of combustion (produced by an increased total supply of air) is obtained by fitting some mechanical means of assisting the draught, which is commonly known as forced draught, and which has already been described. It is really a system for assisting the draught created by the funnel, and can be rendered in two ways: either by (1) increasing the *plenum*, or excess of pressure above that of the atmosphere, by compressing the air before it reaches the firegrate, as in the closed stokehold or closed ashpit systems; or by (2) increasing the *vacuum*, or reducing the pressure inside the funnel still further below that of the atmosphere by increasing the velocity of the ascending column. The differences of pressure, whether of plenum or vacuum, are measured by a water barometer (commonly known as the air-pressure gauge), in which the atmospheric pressure of 14·7 lb. per square inch supports a head of water of 33 feet.

Thus, 1 inch (by gauge) head of water $\Big\} = \dfrac{14{\cdot}7 \times 144}{33 \times 12}$ lb. pressure per 1 sq. ft.

$= 5{\cdot}34$ lb., and the force producing motion

$= 5{\cdot}34 \times A$,

which from (1) $= A \,.\, H \,.\, (w - u)$ evidently.

And, therefore, $H \,.\, (w - u) = 5{\cdot}34$.

$$H = \frac{5{\cdot}34}{w - u} \quad . \quad . \quad . \quad . \quad (6);$$

in which $(w - u) = \dfrac{493\ (T_0 - t_0) \times 0{\cdot}08073}{T_0 \,.\, t_0}$ from (3).

For example, taking the same temperatures as in (5), and substituting the values in (6), then H = 140 feet, about.

That is, under normal conditions the draught created by 1 inch head of water (by gauge) is equivalent to that created by a funnel of 140 feet in height above the grate; and also, under any similar conditions of temperature it should be noticed that the head of water *varies directly* as the equivalent height.

Equivalent Height of Funnel.—From practical experience with a well-stoked fire and with the usual air-supply and funnel-passage areas, it is known that for half an inch of water-pressure, as measured on the gauge, about $12\frac{1}{2}$ lb. of coal can be burnt per 1 sq. foot of grate; and, therefore, in actual practice with efficient

stoking, the pressures should very nearly agree with the rates of combustion given in the following table:—

Rate of Combustion in lb. of Coal per 1 sq. foot of Firegrate.	Total Difference of Pressure in inches of Water.	Equivalent Height of Funnel above Firegrate, in feet.
5	0·2	28
10	0·4	56
15	0·6	84
20	0·8	112
25	1·0	140
30	1·2	168
35	1·4	196
40	1·6	224
45	1·8	250
50	2·0	280

The total difference quoted above is that between the pressure below the fire and that in the base of the funnel; or if two gauges are used, it is the sum of the differences between the atmosphere and the ashpit and between the atmosphere and the base of the funnel. The latter difference is sometimes very small although the height of funnel is great, while the former is very great because so much depends on the stoker in these matters. A well-stoked fire of moderate thickness, say 6 to 8 inches, absorbs from one-half to seven-tenths of the total difference in pressure, which increases with the thickness of the fire. With a fire 24 inches thick, about 5 inches of water-pressure would be required to burn 50 lb. per sq. ft. of grate, the funnel temperature would be about 1400° F., and flaming would be extremely probable; whereas, with a proper thickness, 2 inches of pressure would suffice, the funnel temperature would probably not exceed 700° F., and there would be no probability of flaming.

In warships with a funnel height of 82 feet and upwards, there should be no difficulty in burning 15 to 20 lb. of coal per sq. ft. of grate without mechanical assistance; but in destroyers and other small vessels where the funnel is short—perhaps not more than 20 feet—and high rates of combustion are necessary to produce great power for weight and space, forced draught is imperative. In the latter cases the height of the funnel may be neglected when designing the forcing arrangements, and means should be provided to produce pressures somewhat greater than those shown in the above table for the proposed rate of combustion.

The loss of draught created by funnel damage has already been pointed out, and it thus may be stated that *for all warships* forced draught is necessary, and that there should be a moderately large reserve of power in these forcing arrangements.

Relative Value of Heating Surface.—For water-tube boilers, in which the passage area is nearly constant, or only slightly contracted by baffling, the tubes next the fire absorb about one-third of the total heat given out to the boiler from the fires; the remaining tubes absorb nearly one-half in decreasing proportion, and the funnel and other losses amount to about one-fifth at ordinary rates of combustion. With a greatly accelerated draught the funnel temperature and loss are somewhat increased, and the quantity of heat absorbed by the tubes more remote from the fire might be proportionately increased, because of the greater difference of temperature produced between the gases and the water contained within these tubes; but the quickness of movement of the gases probably causes an actual reduction. For rough practical purposes the following proportions, with a boiler of the Yarrow type, may probably be fairly approximate :—

Rate of Combustion per sq. ft. of Grate.	Percentage of Heat absorbed by each double row of Tubes.							Funnel and other Losses.	Efficiency.
lb.	1st.	2nd.	3rd.	4th.	5th.	6th.	7th.	Per cent.	Per cent.
24 (coal)	30	12	9	8	7	6	5	23	77
36 (coal)	27	11	8	7	6	5	4	32	68
48 (coal)	25	10	7	6	5	4	3	40	60

Furnace Temperatures.—These can only be obtained with any accuracy from careful experiment with proper instruments, but it is possible to obtain a fair approximation by calculation from observations made in previous experiments.

In a furnace the combustion of good Welsh coal produces about 14,400 b.t. units per 1 lb. of coal burnt, and this heat is distributed among about 20 lb. of gases, made up of the products of combustion and the excess of air supplied, with a specific heat of, say, 0·24.

The rise in temperature is in this case about—

$$\frac{14400}{20 \times 0{\cdot}24} = 3000^{\circ} \text{ Fahr.}$$

If only the *minimum* amount, about 11·5 lb. of air is supplied, and complete combustion is obtained, the **maximum** rise is—

$$\frac{14400}{12{\cdot}5 \times 0{\cdot}24} = 4800° \text{ Fahr.}$$

Assuming a fire of about 6 inches in thickness, and that the air is supplied at a temperature of 100° F., the approximate temperatures may be as below :—

Position.	Degrees Fahr.	Remarks.
Below the grate .	100	Air only, not heated before supply.
At 1 inch above .	1600	Excess of air, about 25 lb. per 1 lb. of fuel.
,, 2 inches above .	3000	,, ,, ,, 6 lb. per 1 lb. of fuel.
,, 3 ,, ,, .	4000	Complete combustion practically.
,, 4 ,, ,, .	3000	Lack of air, reducing CO_2 to CO, partially.
,, 5 ,, ,, .	2500	,, ,, ,, CO_2 to CO, completely.
,, 6 ,, ,, .	3100	More air supplied above the fire, burning CO to CO_2.

Heat Lost up the Funnel.—In calculating this quantity it is necessary to know—

1. The heat given out by combustion: say 14,400 b.t. units per 1 lb. of fuel.
2. The temperature of air supplied: say 100°.
3. The weight of air supplied per 1 lb. of fuel: say 19 lb.
4. The temperature of the gases leaving the funnel: say 600°.
5. The specific heat of the escaping gases, *i.e.* the amount of heat contained in 1 lb. by weight: say 0·24 (water = 1).

For example, take the figures stated above; then—

$$\text{Funnel loss} = 20 \times 0{\cdot}24 \times (600 - 100)$$
$$= 2400 \text{ b.t. units per 1 lb. of fuel.}$$

Or, expressed as a fraction of the heat supplied—

$$\frac{2400}{14400} = \frac{1}{6} = 16{\cdot}66 \text{ per cent.}$$

If there be no other boiler loss, such as radiation, then the efficiency of the boiler is—

$$100 - 16{\cdot}66 = 83{\cdot}33 \text{ per cent,}$$

and this must be the *maximum* possible under the conditions stated.

If the escaping gases are used to heat the air supplied for

combustion it is possible to reduce the funnel loss. Mechanical draught must, however, be generally employed, or the draught will not be sufficient to keep the air and gases in motion. The power used to obtain this must also be counted as a loss, but such power when properly applied more than repays the expenditure. The theoretical loss can be reduced to about 5 per cent under exceptional conditions, that is, when the temperature of the air supplied is approximately equal to that of the escaping gases.

The economiser reduces the funnel loss by taking heat from the gases and imparting it to the feed water contained in it.

Howden's system of closed ashpits and hot-air supply, and the Ellis and Eave's system, approach very closely to the highest practical attainment of the least possible funnel loss.

Velocity of Funnel Gases.—These velocities vary considerably in practice for different types of boilers, due to the varying resistances offered by the change in direction and varying passage areas; only an average result can be obtained by calculation, and this is merely an approximation of the actual velocities.

(*a*) The passage-ways allowed in practice for cylindrical boilers are calculated from the grate surface, and are generally a fraction of it; thus—

One-fifth	the grate	area	is allowed	over the fire bridge.
One-sixth	,,	,,	through	the tubes, not including ferrules.
One-seventh	,,	,,	,,	uptake.
One-sixth	,,	,,	,,	armour gratings.
One-eighth	,,	,,	,,	funnel.

(For water-tube boilers, about 5 per cent greater areas are allowed.)

These areas must either be known or assumed; and also—

(*b*) The weight of air supplied per 1 lb. of fuel, say 19 lb.

(*c*) The heat given out respectively to the furnace, combustion chamber, and tubes.

(*d*) The funnel temperature and loss, previously estimated or observed.

If 14,400 b.t. units are given out, experiments tend to show that—

The furnace will absorb about	6000	b.t. units.
combustion chamber will absorb about	3000	,,
tubes will absorb about	3000	,,
funnel loss will be	2400	,,

The heat passing over the bridge = 14,400 − 6000 = 8400 b.t. units, distributed among 20 lb. of gases, of specific heat of about 0·24; therefore—

$$\text{Rise in temperature at bridge} = \frac{8400}{20 \times 0{\cdot}24} = 1750^{c};$$

and if the air is supplied at 100°, then—

$$\text{Temperature at bridge} = 1750 + 100 = 1850^{\circ}\ \text{Fahr.}$$

Similarly, the temperature in the combustion chamber lowered by the heat given out—

$$= \frac{3000}{20 \times 0{\cdot}24} = 625^{\circ};$$

therefore—

$$\text{Temperature at entrance to tubes} = 1850 - 625 = 1225^{\circ}.$$

And again, similarly—

$$\text{Temperature at exit from tubes} = 1225 - 625 = 600^{\circ}.$$

After the temperatures have been found at the various parts, it is possible to estimate fairly accurately the volumes of the gases passing through the passage-ways. But the rate of combustion must also be known. Assume it to be 30 lb. per square foot of grate per hour; then—

Coal burnt per minute $= \frac{30}{60} = 0{\cdot}5$ lb. per square foot of grate.
Weight of gas produced $= 20 \times 0{\cdot}5 = 10$ lb.
And the volume of this gas at 32° Fahr. $= \begin{cases} 10 \times 12{\cdot}38 = 123{\cdot}8 \text{ cubic feet per} \\ \text{minute per square foot of grate} \end{cases}$

(assuming that 12·38 cubic feet go to 1 lb. of gas at 32° F.).

From the law of Charles and Gay-Lussac, $pv = ct$.

$$\text{At the bridge this volume becomes} = 123{\cdot}8 \times \frac{1850 + 461}{493}$$
$$= 578 \text{ cubic feet.}$$
$$\text{At the entrance to the tubes} \quad \text{,,} \quad = 123{\cdot}8 \times \frac{1225 + 461}{493}$$
$$= 421 \text{ cubic feet.}$$
$$\text{At the exit from the tubes} \quad \text{,,} \quad = 123{\cdot}8 \times \frac{625 + 461}{493}$$
$$= 272 \text{ cubic feet.}$$

At the bridge the passage area is one-fifth of the grate area, and—

The **velocity**	over firebridge = 578 × 5 = 2890	ft. per min. =	48	ft. per sec.
„	entering tubes = 421 × 6 = 2426	„ =	40	„
„	exit from tubes = 272 × 7 = 1904	„ =	32	„
„	up funnel = 272 × 8 = 2176	„ =	36	„

If the funnel height and temperature are known, the theoretical velocity of the gases due to these qualities can be estimated; but the resistances reduce this very considerably. The actual velocity in practice is only about one-half the estimated value when natural or funnel draught is used.

Analysis of Flue Gases.—Orsat's apparatus (Fig. 111), of which there are many modifications, consists essentially of (*a*) a graduated

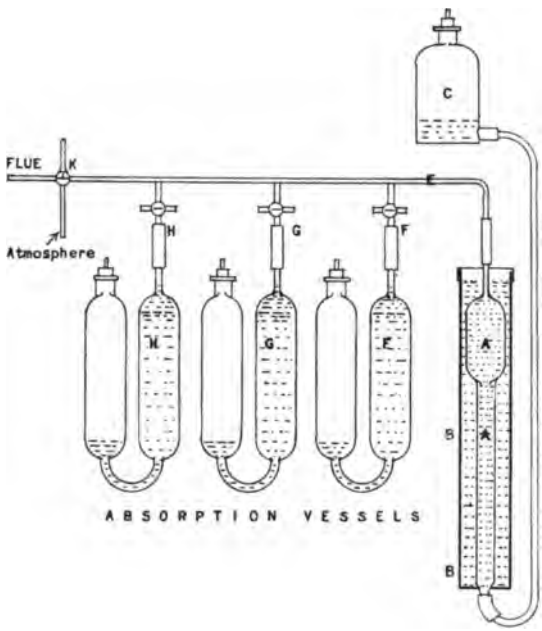


FIG. 111.—Analysis of Flue Gases.

measuring tube *A*, contained within an outer bottle *B*, which forms a water jacket for maintaining a nearly constant temperature; (*b*) a level bottle *C*, filled about two-thirds full with water in the first instance—the top is partially closed, but the bottom is connected through a flexible capillary tube with the bottom of *A*; (*c*) a

glass capillary tube *E*, connecting the top of *A* with three U-shaped absorption vessels, *F*, *G*, and *H*, through stop-cocks, and with the flue gases or with the atmosphere (and a pump) through the three-way cock *K*; (*d*) the vessels *F*, *G*, and *H* are each packed with bundles of glass tubes to increase the absorption surface, and then filled rather more than half-full respectively with a solution of caustic potash of specific gravity 1·20 (for *F*), an alkaline solution of pyrogallol, consisting of the same solution as in *F*, but with from 15 to 25 grammes added of pyrogallol (for *G*), and a concentrated solution of cuprous chloride in hydrochloric acid (for *H*). Copper spirals are introduced into the small glass tubes in *H* to maintain the solution in an unchanged state. The solution in *F* is used to absorb carbon dioxide, in *G* to absorb oxygen, and in *H* to absorb carbon monoxide; and it is important that the gas for analysis should be introduced into the absorption vessels in the order named.

By raising or lowering, the level bottle *C* can be used as a pump, and the method of testing is as follows:—Raise *C* and open *K* to the atmosphere, and thus allow *A* to fill with water up to the capillary part. Shut off *K* from *A*, and open the cock *F*, then by lowering *C* the water runs back into it, and draws up the level of the solution in *F* until it reaches the mark on the capillary neck, when *F* should be shut off. The levels in *G* and *H* are raised in the same manner. The cocks *F*, *G*, and *H* being then closed, *C* is again raised and *A* refilled; then *K* is shut off from *A* and connected with the flue gas-pipe and the atmosphere.

By using a jet or india-rubber pump the flue gas-pipe is exhausted of air and at the same time filled with flue gas. Next, the cock *K* is turned so that the flue gas is in open connection with *A*. This stage of the operations is indicated in the figure. Lower *C* and allow the flue gas to flow into *A* until the water-level reaches zero mark near the bottom of *A*. The measuring tube *A* should now contain exactly 100 cubic centimetres of flue gas, the quantity contained in the capillary *E* being relatively so small that it need not be taken into account. To ensure constant (atmosphere) pressure during measurement, the tube *E* should be connected with the atmosphere after refilling *A*, and after each absorption next described.

Open *F*, and again raise *C*, when the gas flows into *F*, and the carbon dioxide contained in it is absorbed by the solution. Lower *C* until the solution in *F* reaches the former zero (upper) level, and

read off the quantity of gas in *A*. The difference between this amount and the original 100 c.c. is the percentage of CO_2. The process is now repeated to obtain the percentage of O and CO in turn; but the latter is generally very small, may usually be neglected, and is difficult to obtain with accuracy. The remainder volume represents very nearly the percentage of nitrogen. No account is taken of the hydrogen contained in the water vapour of the gases, which with gaseous coals and liquid may be considerable.

For testing boiler efficiencies and for power stations on land, an elaboration of this apparatus is frequently used, and is known as the *Sarco CO_2 Recorder*. By its means the percentage of CO_2 is automatically recorded on a slip of paper revolved on a drum by clockwork past a recording pencil. This arrangement is now adopted for ordinary marine work, and an apparatus is described below. For good Welsh coal the percentage of CO_2 should be *chemically* about 20 per cent; but actually, owing to the necessary excess supply of air, is about 12 per cent. In good practice the maximum is about 15 per cent; above this it points to too little air-supply, the presence of unburnt CO, and possible flaming. The minimum should not be less than 10 per cent; below this the supply of air is too great, and it points to bad stoking with holes in the fire.

Sarco CO_2 Recorder.—The apparatus is preferably hung on a wall or bulkhead, and in the latest arrangement it can be suspended on gimbals for marine use. A $\frac{3}{4}$-inch pipe, in open connection with the flue or base of the funnel, connects with the instrument at 3 (Fig. 112), and in order that the gas samples may be secured rapidly and continuously a similar connection with a point higher up in the flue or funnel is made at 7. The power required to procure and deal with the gas samples is supplied by a continuous stream of water at a head of about two feet.

The water enters at 8 through a glass nozzle 9, of the proper size for regulating the speed of the machine. From 8, the water flows into the tube 74 and power vessel 82, and compresses the air in 82; this pressure is transmitted to vessel 87 through tube 78, and forces the liquid (two parts water and one glycerine) in 87 upwards through tubes 91 and 93. Originally, vessel 87 is filled to mark 95. The liquid is forced upwards through the various branches 48, 49, 51, 66, 67, 68, and 77, and reaches the zero graduated above 67. The moment this head is attained the

syphon 72 begins to flow, and a much larger quantity of water being thus disposed of than flows in through the glass ejector 9, the power vessels 74 and 82 are rapidly emptied; and the pressure in 87 being released, the liquids return from the several vessels and tubes into 87. A partial vacuum is thus created in tube 49, and a sample of gas is drawn into it as the liquid recedes, and this sample rushes into the measuring vessel 68, and some escapes to the atmosphere through 70 and the seal 80. On the next charge, caused by a rise in 82 and fall in 87, the gas in 77 and 68 is forced up into the tube 50, and bubbles through a solution of caustic potash (sp. gravity 1·27) with which the vessel 94 is charged. Any CO_2 contained in the sample is absorbed by the potash, and the remaining constituents of the sample collect in 62, the outlet through 59 being temporarily sealed. This remaining gas collects under two floats, 18 and 26, of which 18 is larger and lighter, and will therefore be raised first. By an adjustment of thumb screws 14 and 15, the stroke of this float is adjusted until just 20 per cent of the whole of the sample (100 c.m.) remains to raise the float 26, *when nothing is absorbed by* 94. If no

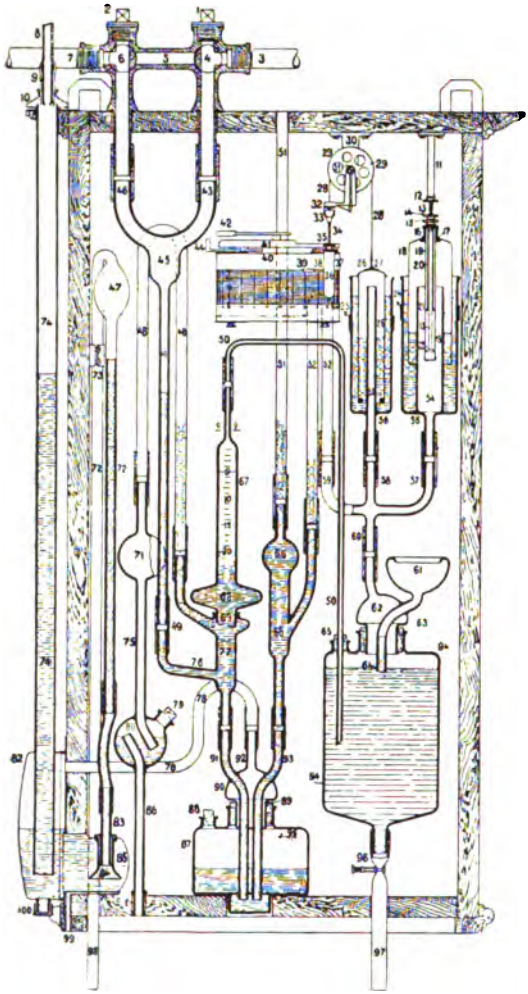


FIG. 112.—Sarco CO_2 Recorder.

CO_2 is contained in the sample nothing is absorbed by the potash in 94, and the whole 20 per cent reaches float 26, and lowers a pen 36, which travels downward on a chart. Any CO_2 contained in the sample is absorbed by the potash, and consequently the travel of the pen is correspondingly limited, and thus the proportion of CO_2 is indicated by the tops of the lines on the chart. On the return stroke of the liquid (87) the charge is withdrawn and expelled through 66 to the atmosphere, and, consequently, a new charge is entirely separated from the old one. The instrument, once erected, works entirely automatically, and requires no attention beyond changing the chart every 24 hours and a renewal of the potash solution every 14 to 20 days. A clockwork mechanism is used to move the chart. As many as 35 to 40 separate analyses can be recorded every hour.

CHAPTER XII

COMBUSTION OF LIQUID FUEL

Flash-Point.—The flashing-point is the temperature at which a mineral oil, when heated slowly in a cup, begins to give off a sufficient quantity of inflammable vapour to burn with a momentary blue flash when a small test flame is brought into contact with the *vapour*.

The general standard used in England is the *Abel closed test*, in which the oil is heated in a closed cup. The flash-point found by the closed test for which a Peusky-Martin tester is frequently used is below that found when the cup is open, and is less subject to inaccuracy of measurement. The flash-point does not refer to the temperature at which the oil burns by means of a lamp wick.

The *point of ignition, firing, or burning* is the temperature at which the *oil* takes fire and continues to burn.

In the mercantile marine fuel oil of about 160° to 200° flash-point has been used for some years, with a practical immunity from accident. The British Admiralty allow the use of oil with a flash-point of 200°; the United States Navy has adopted a minimum of 175°; but a flash-point of 150° to 160° is allowed by the Board of Trade, Lloyds, and other authorities, for use in steel or iron ships. With proper precautions no trouble has been experienced in the safe burning and stowage of oils with a flash-point of 150° F. and upwards.

If the flash-point is comparatively high the fuel oil becomes thick and sluggish, can be burnt with difficulty, and great power is required to atomise it before ignition at the furnace. The volatile elements, which are not present with a high flash-point, are a good assistance to combustion within ordinary limits. The extra refining necessary to produce a high flash-point, by distilling off

the lighter gases, may make the fuel more expensive, unless such lighter products are readily sold for other purposes.

The following table shows approximately the various properties of some of the spirituous oils and fuel oils in common use:—

SPIRITUOUS AND FUEL OILS

Name.	Density at 60° F.	Vapourises at deg. F.	Flash-Point deg. F.	Burning-Point deg. F.	Boiling-Point deg. F.
Alcohol (methylated)	0·83	At any temperature above freezing (32°).	At any temperature above freezing.	About 50 above the flash-point.	173
Petroleum spirit	0·70				136
Benzoline	0·704				139
Benzine, light	0·70 to 0·73				100-158
,, heavy	0·775	...	Above 73	About 125	170 to 180
Naphtha	0·751	...	Above 120	...	...
Kerosene	0·78 to 0·82	...	77 to 122	...	290 to 300
Paraffin		...	73 to 80	...	300 to 572
Russoline	0·824	...	82	...	303
Lighthouse (Broxbourne)	0·81	...	125 to 152	...	329
Lighthouse oil[1]	0·85	...	140	...	300 to 340
Benzol (from coal-tar) (90 per cent Benzol.)	0·88	Any temperature above freezing.	...	...	185
LIQUID FUELS—					
German Navy	0·89	...	150	...	445 to 520
United States	0·89 to 0·92	135	180	210	...
United States	0·91	142	216	240	450 to 600
British Navy	0·9	140	200	220	500 to 600

[1] From a legal standpoint oil is *heavy* when flash-point exceeds 73° F.

COMPARISON OF AMERICAN, RUSSIAN, AND SCOTCH SHALE OILS

	American.		Russian.		Scotch Shale.		Flash-Point deg. F.
	Volume per cent.	Specific Gravity.	Volume per cent.	Specific Gravity.	Volume per cent.	Specific Gravity.	
Benzine, light oils	4	0·700	1	0·725	5	0·730	14
,, heavy oils	2	0·730	3	0·775	...	...	0
Kerosene lighting oils	65	0·810	33	0·822	35	0·805	77 to 122
Intermediate oils	...	...	10	0·858	5	0·850	220
Lubricating pyronaphtha oils	18	0·880	38	0·903	15	0·885	230 to 392
Paraffin wax (vaseline)	2	0·900	...	...	12	0·90	...
Residuum and loss	9	...	15	0·925	28	...	...

Petrol (spirit) contains—

84·3 % carbon, 15·7 % hydrogen, no oxygen, b.t. units per 1 lb. 19,500.

Alcohol (methylated) contains—

41·5 % carbon, 13·0 % hydrogen, 45·5 % oxygen, b.t. units per 1 lb. 11,200.

Methylated spirit evaporates at the rate of nearly ¾ oz. per sq. foot of surface at ordinary atmospheric temperatures, and at about one-third the rate at which petrol evaporates.

Composition and Production.—The elementary analysis of some fuel oils is shown in the following table:—

Source of Supply.	Carbon, per cent.	Hydrogen, per cent.	Sulphur and Oxygen, per cent.	Heat of Combustion (by Calorimeter, b.t. units).	Remarks.
Borneo, crude oil .	87·8	10·78	1·32	18,831	Density, 0·96.
Burmah, residuum	86·4	12·1	1·5	18,864	...
Caucasus, residuum	84·94	13·81	1·25	18,611	Density, 0·94.
Texas, crude oil .	85·66	11·03	3·31	19,242	Density, 0·924.
Beaumont, crude oil	84·6	10·9	4·5	19,060	Flash-point, 180° F. Firing-point, 200° F.
Beaumont oil, refined (U.S.N. tests)	83·26	12·41	4·33	19,481 (estimated)	Flash-point, 216° F. Firing-point, 240° F. Vaporises at 142° F.

If calculated by Dulong's formula, the calorific values would be rather higher; the smaller quantity, obtained by the calorimeter, is probably due to a small amount of water in the fuel.

The crude oil is obtained from wells; but in Scotland, lighting oil is produced by distillation from "shale." Fractional distillation is generally used to obtain the various standard commercial oils and spirits from the crude oil or "shale" oil.

In the early stages of distillation and refining, the lighter gases or oils, with low flash-points, are given off and condensed again ready for storage; but in some cases these gases are used to generate heat for the refining processes.

For American oils generally very little distillation is required, and almost crude oil is used as fuel. This type of oil should therefore be cheap, because the comparative cost of production is low (but the home consumption leaves only a small quantity for export); and in addition the lighter oils are sold for lighting purposes, as well as a great quantity of paraffin wax which remains as a residue. Very little ammoniacal liquor, which is used in the production of agricultural fertilisers, is produced from either American or Russian oils; but from oil shale a large quantity is produced, which reduces the cost of production. Russian oil leaves no wax, but the residue, called mazout or astaki, is used as liquid fuel. Further distillation of astaki results in a large residuum of ozokerit, which is used as electrical insulation.

According to the origin and requirements, governed by the cost of production and market, the crude product undergoes few or many

fractions of distillation, and in this way a large number of spirituous oils—petrol, kerosene, benzine, paraffin, heavy oil (liquid fuel), and heavy lubricating oils of various flash-points and boiling-points—are produced; and in each separate fractional product traces of the various lower flash-point products are to be found. To some extent these different traces, which in the early stages may be a large proportion, cause some difficulty in efficient atomising, and in burning completely, from the early vaporisation of the lower flash-point traces.

As the temperature of distillation increases, the density of the residue also increases, and thus the lighter oils having a flash-point of, say, 73 degrees F. (the English legal minimum), have a low density of, say, 0·68, while the heavier oils with a flash-point of, say, 200 degrees F., have a density of about 0·91; this is seen from the table given above.

Storage.—Refined petroleum, or lamp oil, has very pronounced searching qualities, and its fumes have a great penetrating power, consequently the joint which is quite tight under water-pressure may be subject to leakage with this class of petroleum. As the heavier mazout has a higher flash-point, there is very little difficulty in storing; and a joint which is water-tight is generally tight under the same head of pressure of mazout. A lower flash-point allows some of the more penetrating oils to remain in the fuel, and some trouble might be expected; but when the fuel is stowed in the double-bottom compartments, the lower temperature of the sea-water, which is in contact with a large part of the plating surface, should prevent any excessive formation of vapours and gases, even from the lower flash-point fuels.

The double bottoms are suitable for storage with little alteration, if tested to a sufficient head. Lloyds specify a head of at least 12 feet above the water-line, or the highest point of the tank. They also specify open-topped ventilating pipes to each tank to discharge above the upper deck. Holes should be provided at the highest points of each subdivision of the double bottom, so that the vapour can escape upward from each; while the limber holes, provided for drainage, allow the fluid to gravitate towards the keel. The upper holes would generally be required in the longitudinal frames only, and they should not be large enough to allow any considerable *scending* of the liquid when the ship is rolling. The petroleum vapours are heavy, and tend to accumulate near

the surface of the liquid, and, if the tank is empty, near the bottom of the tank. For the removal of this vapour, which may become dangerous under unfavourable conditions, pump suctions should be led to near both the top and the bottom of the tank. Space must be allowed above the oil for its expansion when warmed by the ship or sea-water outside it.

The open tops of the ventilating pipes should be protected from sparks or lighted matter falling into them.

Settling or Separation Tanks.—The presence of water, which is very common in small or large quantities in the oil fuel, increases the difficulty of obtaining efficient combustion, and considerably reduces the evaporative power. Every 1 per cent of water contained in the fuel reduces the evaporative power by about 2 per cent. The presence of water is generally indicated by the colour of the flame.[1] With satisfactory combustion the flame is an opaque dazzling white for about 6 inches from the end of the jet, then gradually becomes semi-transparent, then at about mid-length turns violet, and towards the end changes to red. When mixed with a moderate amount of water no violet is observed, and the red becomes darkened and fringed with smoke. If only a small proportion of water is present, instead of the dark red, the end of the flame is white, and this is probably due to acetylene.

A large proportion of the sulphur is removed before its delivery for fuel, and the water is dealt with before delivery to the burner.

The densities of the oil fuel (0·9) and water (1) do not differ very much. The lighter oil does not readily rise to the surface, and it is necessary to accelerate the separation by heating or other means. To do this the fuel is pumped into separation tanks, each containing about twelve hours' supply, or a total of about 20 tons for the ordinary cargo steamer. A coil of piping, heated by exhaust steam, is made to float a few inches below the surface of the oil, and the heat sets up a circulation of the oil and water past it. As the oil and water have differing expansive ratios they separate more rapidly when heated. The separation by heating and circulation leaves the lighter oil at the surface, while the heavier water gravitates to the bottom of the tank. A gauge glass shows the relative levels of oil and water, and the water can be drawn

[1] Mr. C. E. L. Orde.

off from the lower part of the tank and the oil left ready for use. The settling or separation tanks are used in rotation.

When separating the oil from water by heating it, only sufficient temperature for the purpose should be allowed in the heating pipe or coil. Vapour is given off by the fuel at a temperature a little below the flash-point, and any great accumulation is not advisable. Exhaust steam, with a temperature not above 140°, is fairly safe for oil with a flash-point of 160°. It is unnecessary and wasteful to heat the whole of the oil in the separation tank, and a pipe led away from the immediate vicinity of the coil is now frequently used to supply the burner. The oil is then supplied to the burner at a high temperature, and is more easily atomised and gasified.

Atomisers, Burners, or Pulverisers.—The usual method of burning oil fuel is to inject it into the furnace through burners of the atomiser or pulveriser type. The object of atomising is to increase the surface area of the fuel by minutely dividing it into spray of the greatest possible number of particles of the smallest size, and thus allow the air required for combustion to become intimately mixed with the fuel. The finer the division is, approaching very closely to the consistency of a fog when using a good atomiser, the greater the surface area becomes, and the more intimate is the mixture with the air supplied for combustion.

The necessary impelling force used for atomising is generally supplied by—

1. Delivering the oil under pressure, as in the mechanical spray type. (Similar in principle to a fire-hose.)
2. Inducing a flow by a jet of steam, which is frequently superheated. (Similar in principle to a boiler feed-water injector.)
3. Inducing a flow by a jet of compressed air, which is frequently heated.
4. By gasifying the oil in a heater (or vaporiser) and allowing it to escape through the burner.

Any one of these methods can be used either for oil fuel combustion only, or for mixed combustion, when the oil fuel is injected on to the incandescent coal. Various methods are adopted for increasing the efficiency of the combustion: such as heating the air supplied for atomising or for combustion, heating the fuel, superheating the steam used for atomising, and in many cases a combination of some or all of these is employed.

When the firegrate is retained for mixed combustion the burners are generally fitted near the furnace doors, and in some cases a burner is attached to each door. The burners point slightly downwards, so as to direct the jets on the middle of the fire or thereabouts.

When the firegrate is removed the arrangements vary considerably. The usual way is to fit the burners as low as possible in the furnace, commensurate with the proper supply of air below the burners, and with obtaining as much room as possible for the completion of the combustion before the hot gases reach the boiler heating surfaces. If the hot gases strike the boiler heating surfaces before combustion is complete, and are cooled by it to a temperature below that necessary for combustion, the combustion is checked and the gases escape only partly consumed, and smoke is produced. In all cases a sufficiency of brickwork is necessary to perfect combustion: it retains the temperature of the furnace at a higher level, and, acting as a heat-accumulator, re-ignites the gases and unconsumed oil, and it also tends to keep the flames in a state of incandescence.

Furnaces, Screens, and Brickwork.—The arrangement of the furnace is of the greatest importance to efficient combustion, and the whole of the available space should be utilised.

When the temperatures of the combustible and the air supplied for combustion are low, the flame is long; but when these temperatures are sufficiently high, the flame can be shortened to the available length of the furnace, and the time required to complete combustion is also shortened.

The impact of the flame on a cold surface, such as a boiler-plate or water-tube with a temperature of about 400°, checks combustion and produces smoke, as the petroleum deposited does not readily re-ignite. It was usual, therefore, to direct the jet on to a brickwork erection, which, retaining some heat, re-ignited the spray falling on it, and burnt the petroleum away. If the furnace is sufficiently long, it is preferable to leave the flame unrestricted (at present a common practice), but sufficient brickwork must exist to ensure the re-ignition of any fuel dropping on the masonry.

For burning oil only, the furnace arrangement shown in Fig. 113 has been fitted by the Wallsend Slipway and Engineering Company. The ss. *Murex*, fitted with a somewhat similar arrange-

ment, accomplished a trip from Singapore to the Thames, *via* the Cape, without touching the furnaces, and required an oil fuel consumption of only 16 tons per day, instead of the previous coal consumption of 25 tons per day. The oil is sprayed, through Rusden-Eeles burners, by steam on to the brickwork baffle *A*. The flames and hot gases then pass over the top and around the sides of the baffle into the second part of the furnace, where combustion continues from the further mingling of the gases. The throat of the furnace is fitted with another baffle *B*, which prevents

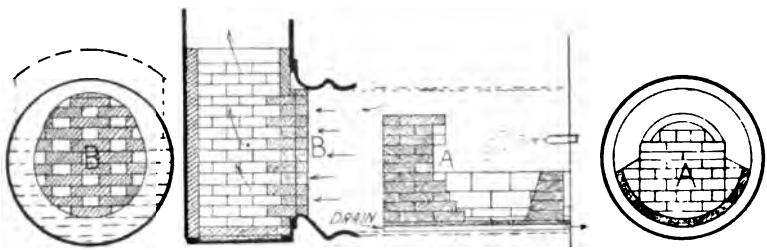


FIG. 113.—Furnace for burning Oil only (*Murex*).

the direct impact of the flames on the rivets and stays in the combustion chamber.

A trough is formed by brickwork below the burner, and any oil which may drip from the burner is re-ignited by the heat of the surrounding brickwork. A drain-hole is formed between the brickwork and bottom of the furnace for the escape of any water which may be deposited behind the baffle *A*. The throat of the furnace is built up with bricks, leaving passages for the gases, as shown in the left hand sketch. The lower part of the combustion chamber is lined with brick at sides and bottom, and is thus protected from any direct impact of the flames.

When the firegrate is retained, the arrangement shown in Fig. 114, fitted to the ss. *Trocas*, is sometimes adopted, and can be used for either mixed or separate combustion of coal and oil. When burning oil only, the firegrate is covered with broken firebrick. A brickwork arch protects the throat of the furnace at the upper joint with the tube plate, and the back of the combustion chamber is also protected by brickwork. The furnace front shows the arrangement of two Rusden-Eeles burners, which can be swung round pivots clear of the fire-door when coal is being used. Although in this case coal firing can be quickly changed for oil,

it is preferable to remove the firegrate and fit up the furnace with the necessary brickwork when burning oil only. Improved efficiency is obtained by using the greater available space for combustion, and little time is required for effecting the change.

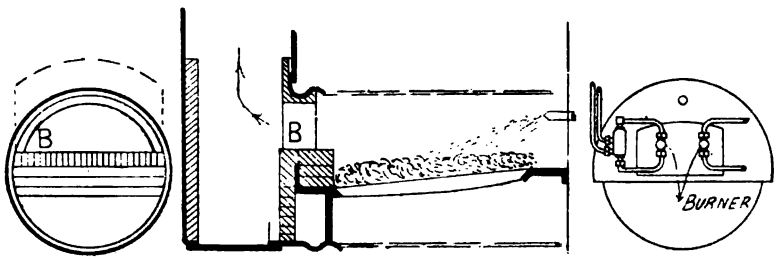


FIG. 114.—Furnace for burning mixed Coal and Oil.

The arrangements used in warships for mixed coal and oil combustion are shown in Figs. 81 and 115, for a Yarrow boiler and a cylindrical boiler respectively. Taking Fig. 115 as an example, two burners are fitted, one on each side of the furnace door. Air is supplied from the stokehold through holes in the circumference of a

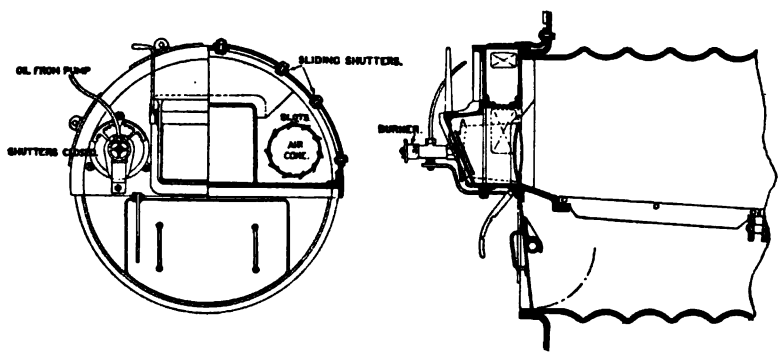


FIG. 115.—Furnace for burning mixed Coal and Oil.

projection from the end of the furnace, and the amount is regulated by sliding shutters. The burner sprays the oil into the central space of an air cone, with slots cut in its conical face. The slots are made so that the air, entering through the sliding shutters and the slots, is given a centrifugal spiral motion in an opposite direction to the centrifugal motion of the spray, which is thus very finely

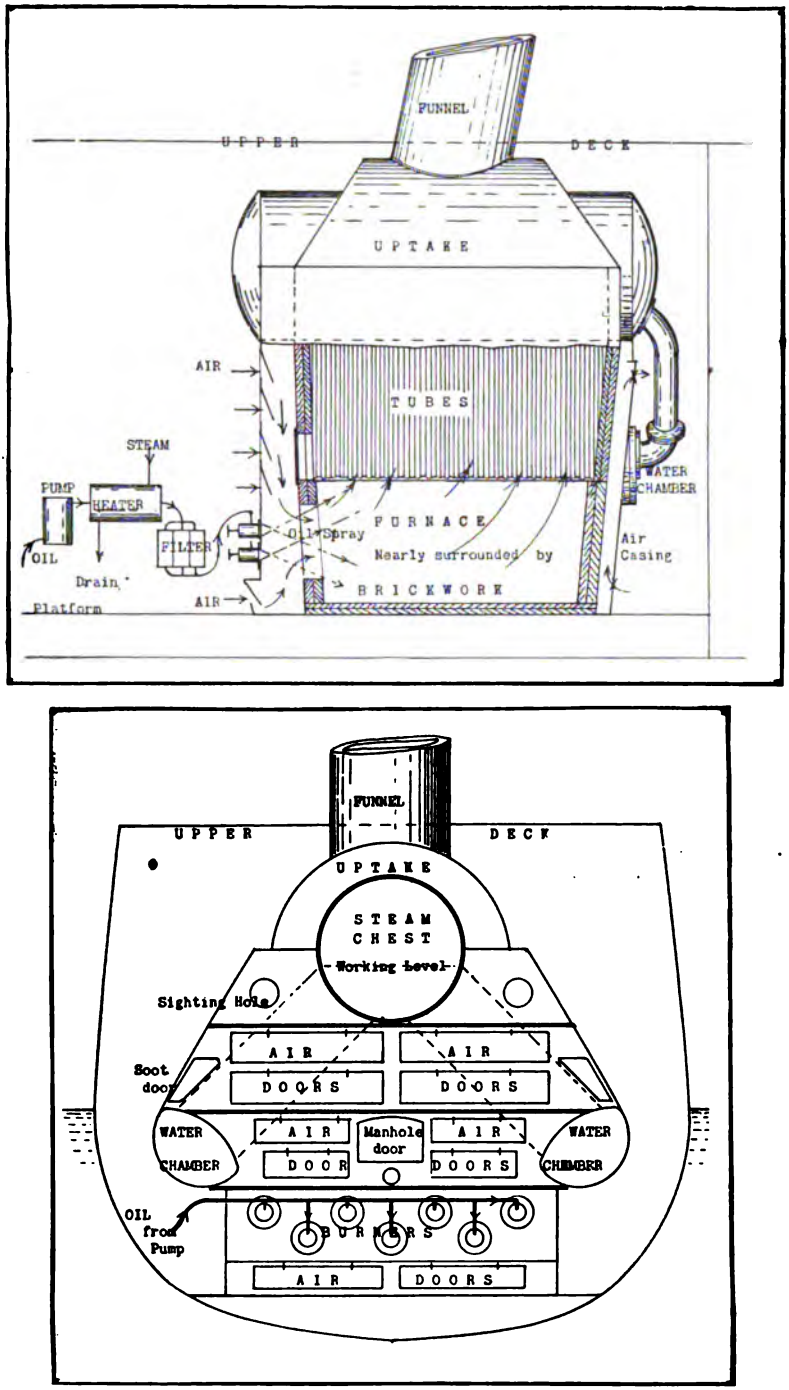


FIG. 116.—Arrangement of Oil Burners, Water-tube Boiler.

divided, and burns completely when the air and oil supply are properly regulated. Air is supplied to the coal fuel through the ashpit in the usual manner. The position and general arrangement of the burners, air cones, sliding shutters, etc., are clearly shown in the figure, and when seen the various details can readily be followed.

A general arrangement of a White-Forster boiler using oil fuel only, as fitted in recent destroyers and torpedo boats, is shown in Fig. 116. No firegrate is fitted, and the furnace is surrounded by brickwork except on the sides which are flanked by the water-tubes. Air is admitted through a number of air doors on the self-closing principle, and the stokehold is under forced draught (plenum system). A large number of burners are fitted—seven, as shown. An outline of the system is also shown, with the pump, oil heater, filter, and burners. At the back end of the boiler, behind the brickwork, there is an air casing with natural circulation caused by convection between the inner casing, against which the brickwork rests, and the outer casing. A manhole plug is fitted for access to the furnace for repairs and examination; and sight holes are fitted front and back, so that by fixing a lamp at the back end the condition of the gases, whether smokeless or otherwise, can be seen. The sight holes are fitted with coloured glass and require periodical cleaning, so that the gaseous condition can be properly estimated.

The general arrangement for burning oil fuel, either in conjunction with coal, or alone, in a boiler is shown in Fig. 117. The oil enters the oil fuel pump, *Q*, through a suction pipe and valve *D*, and is delivered under pressure through a valve *E*, and air vessel (of the type shown in Fig. 201), into one or both of the filters (cold) *G*, which are arranged so that either one can be cleaned while the other is in use. From the filter, *G*, the oil enters the oil heater (Fig. 122), which is of the surface type, and warmed preferably by exhaust steam from the closed (auxiliary) exhaust. The oil when warmed is thinned and flows more easily, and is again filtered through filters *J* (hot), before passing into the burners through valves *K*, *L*, and *R*. *M* is a separator on the heater (steam or water) drain, and *O* is the separator drain valve; when working steadily the highest economy of heating steam is obtained when the drain *O* is set so as to keep the separator about half-full of water, as shown by the water-gauge. Pressure-gauges on the oil delivery system show the necessity or otherwise of cleaning the filters, and a

difference of 5 lb. and 10 lb. per square inch respectively between the air vessel and cold filter, and between the cold and hot filters, generally obtains when the system is in good working order; the precise pressure difference is, however, a matter for observation in practice under working condition with each plant.

Starting the Burners.—When compressed air is used for atomising, a certain amount can be stored for future use, or a hand

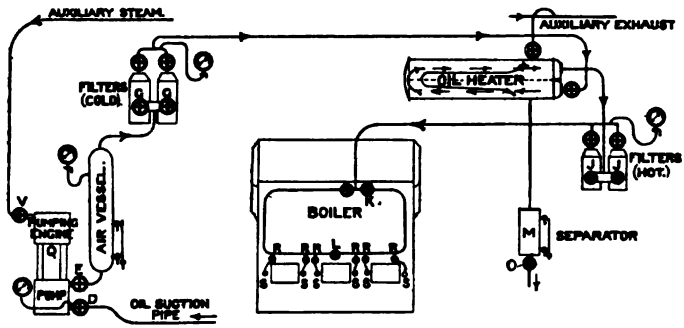


FIG. 117.— Arrangement for burning mixed Coal and Oil.

compressor can be used until sufficient combustion is started or sufficient steam generated to work the pump.

In the ss. *Mariposa* steam is first raised by burning coal in an auxiliary boiler to start the air compressor, heat the oil, and set the burners in operation.

In some cases the oil is led through a coil of copper piping, which can be heated by lighting some oily waste under or about it. This warms the oil fuel and gasifies some portion of it. It is then sprayed at the nozzle, and directed on to a piece of lighted oily waste placed on the firegrate.

In the Kermode system, with hot air, the furnace doors are fitted with burners, and lighted waste is thrown on the broken fire-brick, or asbestos, which covers the firegrate. A few strokes of the pump sets the jet to work, and it then continues automatically as the oil is allowed to gravitate from a tank to the burner.

Mechanical Spray Burner.—Two burners of the Körting type are shown in Fig. 118. The oil is raised from a separating tank by a steam-driven pump, and on its passage through the suction pipe is heated to a temperature of about 150°, and filtered before reaching the pump suction valve. It is then forced by the pump

through the delivery pipe, which passes through the hot-air casing of the boiler, and is further heated to about 200° to 270°. The fuel is again filtered before delivery through the burner into the furnace, and is sprayed under a pressure of about 30 to 50 lb. per square inch.

The area of delivery is controlled by a central spindle, which is fitted with an enlargement of spiral form. The passage of the oil past the spiral imparts a vortex motion to the issuing fluid of sufficient intensity to break it into spray. The spraying is sometimes increased by jets of air, which are entrained through the holes in the flange at the end of the nozzle; the jets of air strike

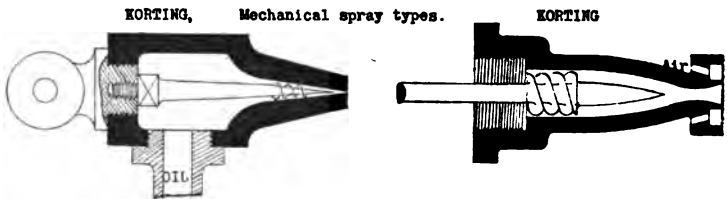


FIG. 118.—Körting Burners.

the vortex of spray at an angle, and tend to still further divide it or atomise it.

The air supplied for combustion is heated by the Howden system in some instances, and then the combination of heated fuel and heated air produces a very efficient combustion.

The weight or quantity of oil discharged through a mechanical spray burner or nozzle varies as the square root of the pressure or head, because it must be in general agreement with the usual formula—

$$v^2 = 2 \,.\, gh.$$

If p = pressure per square inch of oil just previous to discharge, then $v = 8\sqrt{2 \,.\, p}$ about, if something be allowed for frictional losses.

On this basis the area A in square inches, and diameter D in inches, of the nozzle bore can be found fairly accurately for a given set of conditions, which is best shown by an example.

EXAMPLE.—The total consumption of fuel oil per hour is 2·59 tons (= 100 cubic feet about), and there are 25 nozzles; find the diameter of the nozzle

bore for oil-pressures of 8, 40½, 128, and 312½ lb. per square inch respectively.

Cubic feet of oil discharged per nozzle per second =

$$\frac{A \times v}{144} = \frac{4}{60 \times 60}$$

$$A = \frac{4 \times 144}{60 \times 60 \times 8\sqrt{2p}}$$

$$= \frac{1}{50\sqrt{2p}}$$

$$D = \sqrt[2]{\frac{A}{\pi}} = \frac{7}{4}\sqrt{\frac{1}{50\sqrt{2p}}} = \frac{1}{4}\sqrt[4]{\frac{1}{2p}} \text{ about } \left(\text{assuming } \sqrt{\frac{1}{\pi}} = \frac{7}{4}\right).$$

If

$p = 8$, then $D = \frac{1}{8}$
$p = 40\frac{1}{2}$, then $D = \frac{1}{12}$
$p = 128$, then $D = \frac{1}{16}$
$p = 312\frac{1}{2}$, then $D = \frac{1}{20}$

The diameter in practice is seldom greater than $\frac{1}{20}$-inch, and although in the above example as great a diameter as $\frac{1}{8}$ might be used on a calculation basis with a low pressure, the efficient atomising of a stream greater than about $\frac{1}{16}$-inch is very problematical, and would require in practice a very great, not a very low pressure.

Thus, for general purposes of calculation, it is perhaps better to start with a fixed idea of the nozzle area, and from the weight (or cubic feet) of oil to be burnt per hour to calculate the number of nozzles.

A burner naturally has a particular rate at which it is most efficient, and although some moderate variation may be possible with a fair degree of efficiency, it is evident that a variation of power of the boiler is best attainable by regulating the number of nozzles in use; but it should not be forgotten that it is extremely difficult to regulate a great number of nozzles so that any one of them shall not produce smoke. Thus actually the number of nozzles is as few as possible, and contingent on each one being worked at full power with the highest practicable efficiency.

Steam Atomiser.—The objections to the use of steam for atomising in marine boilers are:—

(*a*) About 3 to 5 per cent of the steam generated in the boiler must be used for atomising, and the fresh feed water so lost must be made up by the evaporating plant. Usually extra plant must be supplied for this special purpose.

(*b*) Unless the steam is sufficiently superheated (or the oil heated, or the air supplied for combustion heated), it becomes condensed when ejected from the nozzle, and tends to reduce or put out the incandescent flame, and to produce smoke.

(*c*) It does not assist in the combustion of the fuel, and all the heat contained in it is wasted by passing up the funnel as steam (containing both latent heat and superheat).

When the supply of fresh water is unlimited, as in general railway practice, the steam atomiser is perhaps the most suitable, and good results have been obtained both in combination with coal and when oil only is used. In the Holden system the air-supply pipe is led through the smoke box and heated before it reaches the burner; the oil is also heated, and sometimes the steam is superheated. All these contribute to efficiency.

The *Holden burner* is arranged for a central jet of air, enclosed in a cone supplying the steam which forms the atomising agent. Outside the steam-supply there is another concentric cone containing the oil fuel. The three supplies—air, steam, and oil—meet at the ends of the cones, and spray together, through the nozzle, into the furnace. The breaking into spray is further increased by a ring of steam jets converging on the mixture issuing from the nozzle. The result is a more intimate mixture of the oil and air, and a more general diffusion of the flame of combustion inside the furnace. The principal air-supply enters from below the jets, and is regulated to produce smokeless combustion.

Texas oil produces about twice the weight of steam, compared pound for pound with coal, as used for locomotives on the G.E.R.

When steam is used for atomising, a separator is generally fitted in the steam pipe leading to the burner.

The Kermode burner, shown in Fig. 119, is used with steam as the atomising agent; but the Kermode system is generally used with hot air and with compressed air as the atomising agent. The burners used for steam and for compressed air do not differ materially in principle. The oil and steam enter through the pipes shown, each of which is fitted with a regulating valve. The oil passing through the central valve is given a right-hand twirl by the stem of the valve. The steam, forming the inducing agent, surrounds the oil cone containing the valve stem. The air for combustion enters through the holes shown, and the supply can be regulated by moving the perforated band shown in the right-hand figure. This air, on

its passage between the steam-containing cone and the outer case, is given a left-hand twirl, and, meeting the oil, mixes intimately with it during the passage to the furnace and just inside the furnace. The supply of steam can be regulated by adjusting the two parts of the burner, which are screwed together, as shown. The front end

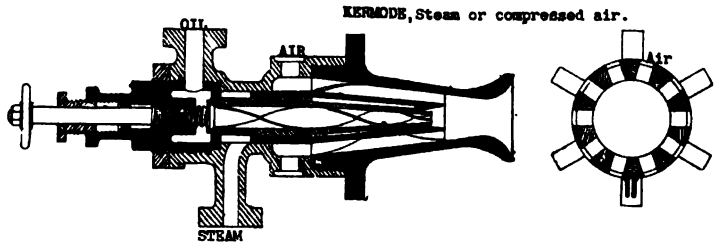


FIG. 119.—Kermode Burner.

of the burner moves the air cone, which is attached to it, and so regulates the opening between the ends of the oil and air cones.

Belleville Burner.—A type of burner fitted in connection with Belleville and other boilers is shown in Fig. 120. The oil connection is fitted in front of the gas mixer, and opposite it is the jet sprayed outward by the movement of the air under pressure in the gas mixer through a coned space formed by a valve-seating. This

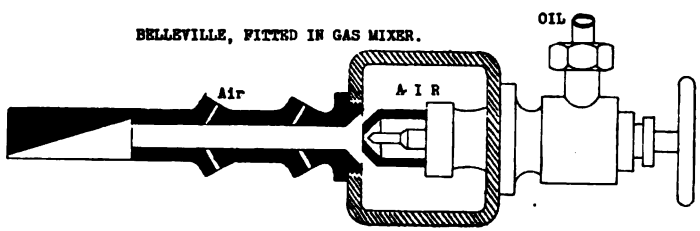


FIG. 120.—Belleville Burner.

arrangement has been used with steam as the atomising agent, but is more suited to air, with which for moderate rates of combustion it is fairly satisfactory.

Compressed-air Atomiser.—Nearly all the steam atomisers can be adapted for use with compressed air as the atomising agent.

In the Kermode hot-air system the burner is very similar to the one already described above. The supply of air for combustion passes through a pipe from the front of the furnace, along one side,

then across the back, and, returning by the opposite side of the furnace, meets the oil fuel, which is also heated to a high temperature. The oil on its way to the burner passes through a vaporising pipe fitted inside the furnace. The objection to the system, as described, is that the supply of heat for both the oil and air is taken from the furnace, and not from the waste heat in the uptake or other part of the plant. Increased efficiency is undoubtedly obtained by the better combustion from heating the oil and air, but the same result could be more economically obtained by utilising waste heat for the purpose.

Experiments were made by a United States Navy Board with a water-tube boiler of the Hohenstein type, with tubes nearly horizontally arranged. The most successful type of burner

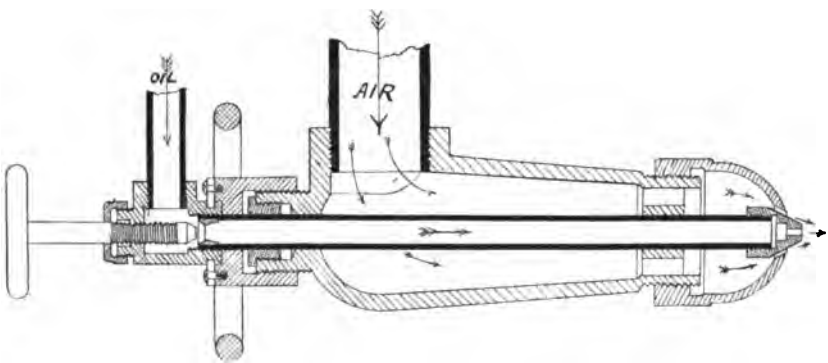


FIG. 121.—Oil City Boiler Works Burner.

is shown in Fig. 121, and was supplied by the Oil City Boiler Works. The boiler had previously given satisfactory results with coal, but it is not considered necessary to describe it. The air for combustion was supplied under pressures varying from 0·78 to 4·68 inches, and averaged about $3\frac{1}{2}$ inches of water, by a Root blower. A special system of mirrors was arranged so that the stoker could see the smoke issuing from the funnel. By giving the burner an excess of oil smoke was produced, and the supply was then gradually reduced until the smoke became very faint. The results, although the best of their date, have been entirely outclassed by recent results in the Royal Navy; but a study of the subject-matter of the report is of considerable value, and throws a good light on many obscure side-issues.

Oil Fuel Heater.—It has already been pointed out that when

the oil is heavy it atomises very badly at a low temperature, but by heating it to a moderately high temperature the oil becomes thinner and, running freely, atomises easily. For high rates of combustion an oil heater is necessary to good combustion, and one is shown in Fig. 122 as fitted in a torpedo boat burning oil fuel only. The heater is of the horizontal surface type, and the oil circulates through the upper tubes and returns through the lower rows. Between the upper and lower rows of tubes a diaphragm plate is fitted, so that the steam when condensed generally fills the lower space, access to which is given through $\frac{1}{4}$-inch holes in the diaphragm plate, about 12 at the steam inlet end *A*, and about 2 at the outlet end *B*. A drain conveys away the condensed water as

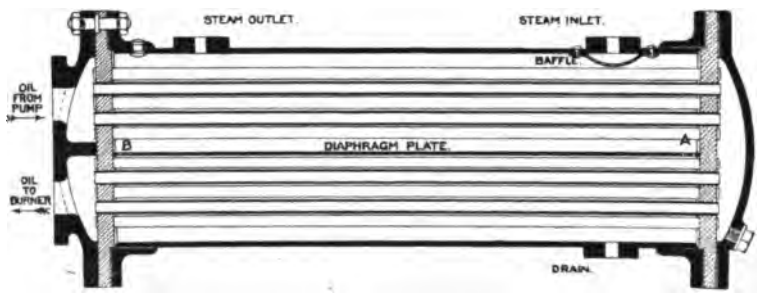


FIG. 122.—Oil Fuel Heater.

required. It is preferable for the oil to pass through the tubes internally, so that the surfaces can be more easily cleaned by a tube brush.

Oil Fuel Filter.—To prevent gritty matter choking the burner openings, which are very small, the oil is filtered after being heated and before it reaches the burners. Strainers of 36 and 24 meshes to the inch of wire gauze are fitted respectively in series on an outer and inner cylinder, as shown in Fig. 123, and can easily be removed and cleaned.

Water evaporated per 1 lb. of Fuel.—The combustion of 1 lb. of the best Welsh coal will, theoretically, produce about 15 lb. weight of steam; but in practice, only 8 to 12 lb. are produced. From many practical experiments with oil fuel, it may be taken that from 13 to 18 lb. weight of steam can be produced by the combustion of 1 lb. of fuel oil.

The relative values of oil and coal, based on their practical

evaporative power, are probably in the ratio of about 5 to 3. This ratio is confirmed by Colonel N. Soliani, in an Italian torpedo boat; by experiments with the French cruiser *Surcouf*, fitted with Belleville boilers, and by the trials of the *Gueydon*, fitted with Niclausse boilers; and by the results obtained by the ss. *Murex* (*Transactions of the Institution of Naval Architects:* Paper by Sir F. Flannery).

The efficiency of evaporation depends on the rate of evaporation, and the rate of evaporation must depend on the rate of combustion, which affects both coal and oil fuels. The efficiency of combustion of oil fuel falls off less quickly than that of coal, because the

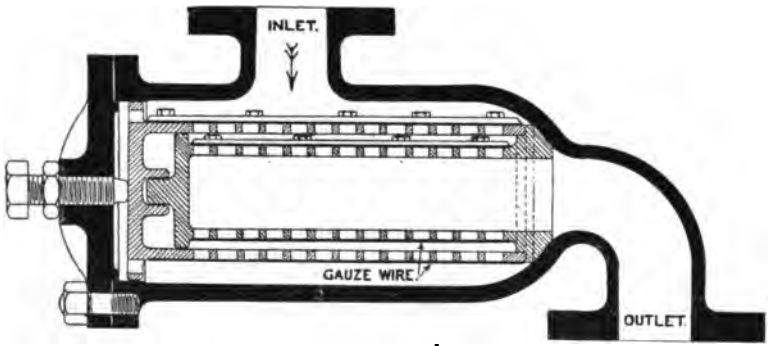


FIG. 123.—Oil Fuel Filter.

furnaces and tubes usually remain clean, which constitutes an advantage.

Rate of Combustion.—In the mercantile marine, where the rate of combustion is seldom greater than from 15 to 20 lb. of fuel oil, there is no difficulty in obtaining almost perfect combustion without the production of smoke. The early destroyers required a rate of combustion of from 60 to 90 lb. of coal per square foot of grate to obtain their full power. In the later types the coal consumption is limited per I.H.P., and the rate of combustion has been decreased to obtain the necessary increase in efficiency. The coal consumption is thus reduced to about 45 or 65 lb. per square foot of grate. The relative oil fuel consumption would then be about 27 to 30 lb. It is usually when trying to burn at, or above, this rate that the combustion of oil fuel is unsatisfactory, and large volumes of smoke are produced.

The rate of combustion which can be efficiently used with any

particular boiler depends principally on the volumetric space allowed, and other necessary details.

Relative Economy of Coal and Oil Fuel.—An interesting comparison, in addition to those given above, is afforded by the trial reports of some torpedo boats built by Messrs. J. S. White of Cowes, equipped respectively with piston engines in conjunction with coal-fired boilers, and with Parsons turbines in conjunction with oil-fired boilers. The boilers in each case are of the White-Forster water-tube type.

	Piston Engines and Coal.	Turbine Engines and Oil Fuel.
Tonnage displacement . .	205	245
Number of boats	4	5
Average I.H.P.	3000 about	4000 (estimated)
Average mean speed on trial .	24·824 knots for four hours.	26·36 knots for eight hours.
Consumption of fuel per hour .	2·73 tons	2·59 tons
,, ,, ,, mile .	248 lb.	220 lb.
Distance run per ton of fuel at about 12 knots . . .	...	37·90 nautical miles

From the table it will be noticed that, apart from the advantage which is undoubtedly gained by using turbine engines, there is a gain of about $1\frac{1}{2}$ knots (obtained in Stokes Bay, and therefore, probably, a little below that obtainable under similar conditions but in deeper water); the full power can be maintained with oil for so long as the fuel lasts, as opposed to a maximum period of about four hours with coal; and the consumption per mile is rather over 11 per cent greater with coal than oil, although the displacement on trial was 30 tons greater with the turbine and oil fuel installation.

Requirements for Good Combustion. — The space, both horizontally and vertically, required to ensure good combustion is closely allied to the rate at which the fuel can be burned. If the space cannot be increased—by lowering the furnace floor, for instance —then the time must be taken into account. The heating of gases and air supplied for combustion requires a certain time, and if the heating is effected in the furnace itself, the rate of combustion is proportionately decreased. By heating the air-supply as much as possible before it enters the furnace, and by heating the fuel-supply so as to gasify a sufficient amount of it before spraying and ignition,

the time that is necessary to retain the combustion gases in the furnace can be decreased, and thus allow an efficient increase in the rate of combustion.

If the oil is heated before supply to the burner, the limiting temperature is about that of the flash-point of the fuel; but if the fuel is heated in the burner itself, or very close to it, the limit can with safety be much higher.

If the air supplied for combustion is heated by the waste funnel gases, the limiting temperature is simply the greatest which can be obtained in this way. Every unit of heat so extracted is an economical gain, and, in addition, a better efficiency of combustion is obtained, with a higher temperature in the furnace and a nearer approach to the minimum quantity of air required for combustion. A smaller supply of air means also a smaller funnel gas velocity, and an increased time for the passage of heat through the boiler heating surfaces to the water inside the boiler.

The oil should be filtered or strained and separated from water, and its temperature should be sufficiently high so as not to produce any condensation of the steam used for atomising; and if superheated steam is used for atomising, the superheat should be sufficient to prevent condensation when meeting the fuel and air supplied for combustion.

The surrounding brickwork and baffles must retain sufficient heat to keep the sprayed fuel in an incandescent state, and to complete the combustion before the heated gases are reduced in temperature by contact with the boiler heating surface. There should be equal distribution of the fuel over the whole of the available furnace area, with a sufficient pressure in the jets to reach all parts.

If high rates of combustion are required, it is probable that hot compressed air is a more efficient atomiser than either saturated or superheated steam, because the air supplies oxygen for combustion, while the steam, displacing the air, reduces the available space for combustion; but the mechanical system has given the best results over a considerable range of rates of combustion.

Advantages claimed over Coal.—Assuming the same weight to be carried in each case, the advantages of oil fuel may be considered as below:—

1. Greater energy available in the fuel; increase about 25 to 30 per cent.

2. Extension of steaming distance; increase about 15 per cent.

3. Reduction in space for stowage; decrease about 10 per cent.

4. Ease of transhipment and stowage, probably at the rate of 300 to 400 tons per hour through a very short length of pipe.

5. Reduction in number of firemen; decrease about 50 per cent.

6. Maximum speed at short notice, with continued efficiency, because there are no fires to clean. If the combustion is not smokeless, tubes must be swept at intervals.

7. Less wastage at varying powers, because the fuel-supply can be shut off at once from the furnaces, or varied to suit the power required. With coal, the furnaces cannot be cleared of the coal in them, which roughly amounts to about one ton for each water-tube boiler, and this amount is necessarily wasted when the ship is stopped.

PART IV.

THE MARINE STEAM ENGINE (PISTON TYPE)

CHAPTER XIII

MODERN FORMS OF MARINE ENGINES

Vertical Engine and its Advantages.—For marine machinery generally, with the exception of the recent adoption of the steam turbine for naval purposes, the *vertical inverted* direct-acting engine, as shown in Fig. 124, has been adopted, and its advantages over previous types are :—

1. The ground space occupied by it is much smaller.
2. Greater accessibility for erection and repair.
3. Less friction and wear on all the working surfaces of the bearings and reciprocating pieces of the mechanism.
4. For merchant vessels and in warships, where the height is sufficient, a longer stroke can be used. This has some advantage, as it reduces the loss of steam and water from leakage at the glands of the piston and slide rods; with a long stroke a slower rate of revolution of the propeller can be used, which generally produces a greater propeller efficiency, and there are some other incidental advantages.

Since about 1888 very few engines have been built, even for the Navy, where they have not been of the vertical type. In a few cases, previous to 1889, where the engines could not be kept below the water-line, they were protected by an armoured citadel, and several cruisers were fitted in this way; but about 1895-96 the stroke of the engines for these small vessels, following on the lines of destroyers' engines, was decreased, and with it the height was

also correspondingly decreased. This decrease of height will be better appreciated by studying the section on "height of engine"

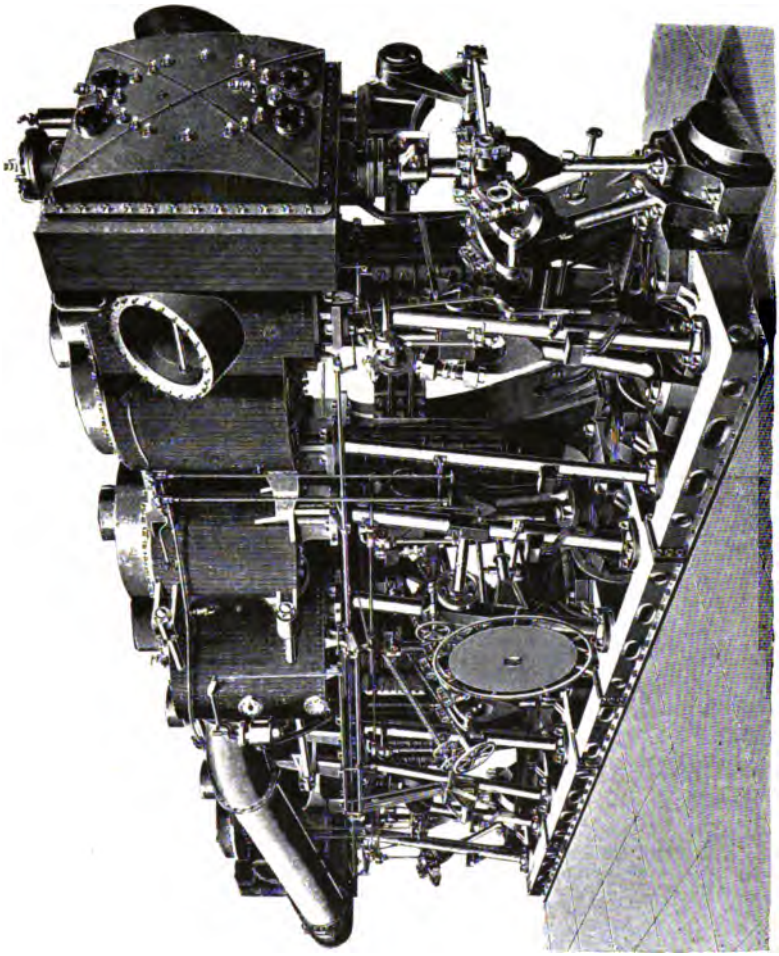


FIG. 124.—One Engine of H.M.S. *Black Prince*.

and the "table of various details" given later. Unless otherwise distinctly stated, the vertical inverted engine is always considered in this book.

Necessity for a Guide.—In Fig. 125 diagrams are shown of a vertical engine in the relative positions for the up and down strokes. When the engine is going "ahead" the pressure on the guide *G* is in the *same direction*, whether the piston *D* is either

pushing the rods *E* and *R* downward or pulling the rods upward; and so long as the shaft continues to move in the same direction of rotation *under the influence of the piston*, only the one guide shown is necessary to keep the piston rod in line with the axis of the cylinder *A* and piston *D*.

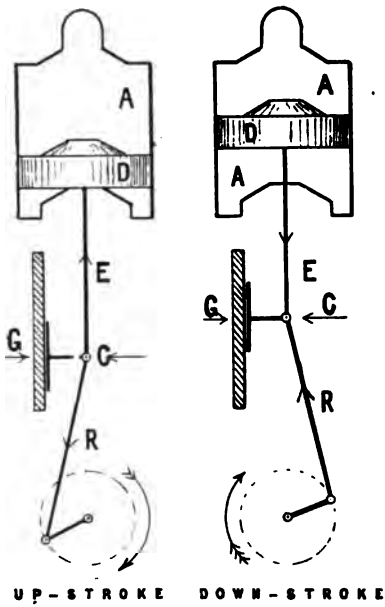


FIG. 125.—Necessity for a Guide.

If the direction of motion of the shaft be reversed, to go "astern," a guide is necessary on the opposite side of the crosshead to take the pressure in the opposite direction. This arrangement is fitted in many mercantile engines; but for naval purposes a lip is formed in connection with the guide used for going ahead, and embraces projections on the crosshead so as to take the astern pressure on the guide in the opposite direction. This arrangement is clearly shown in Figs. 124, 127, 128.

The *direction of rotation* of the shafts when going "ahead" varies in a twin-screw ship correspondingly with "inward" and "outward" turning screws. For inward turning, fitted in many naval vessels between 1895 and 1902, the *port* shaft revolves, as shown in Fig. 125, when going ahead as viewed from aft, while the starboard shaft revolves in the opposite direction; the cranks when above the shaft centre turn *towards* each other. With outward turning screws, the cranks when going ahead and above the shaft centre turn *away* from each other; and the back frames, to which the guides are attached, are back to back with the mid-line bulkhead between them. (See also Chapter XXV., "Inward and Outward Turning Screws.")

Obliquity of the Connecting Rod.—It should be noticed that when the piston is at mid-stroke *M*, the crank is not exactly at right angles to the line of stroke (produced). In Fig. 126 *A* and *B* are the top and bottom ends of the stroke of the crosshead, which exactly agrees with the relative position of the piston in its

stroke because they are rigidly connected. The length of the connecting rod = $OM = AN = BK = MR$, as shown in the figure. When the crank is at OP, at right angles to AO, the position of the crosshead is at C, where $PC = MO$, and C is not coincident with M but a little below it. When the crosshead is at mid-stroke M, the crank is at R (or in a similar position on the opposite side of AO), and the position of the crank relative to AO is at L, where LR is at right angles to AO. If the rod is of infinite length, then C becomes coincident with M, and L with O, and the relative motion is harmonic. There is a modification in which the cylinder is placed *not* immediately over the crank. (See Chapter XXXV.)

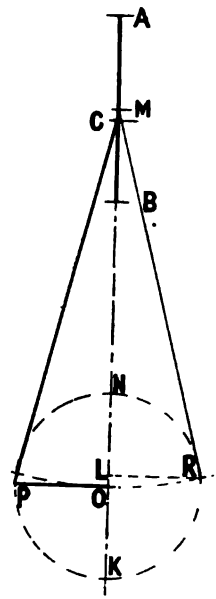


FIG. 126.—Obliquity of Connecting Rod.

The error, so called, due to obliquity is generally neglected in elementary calculations relative to the connecting rod, but is of importance in the accurate design and setting of the slide valve, as well as in closely approximate calculations of the various forces and turning moments acting on the crank.

The Engine Framework.—In Fig. 127, and in the reference table below, the various names and positions of the principal parts of a vertical inverted engine are stated. The figure shows only one (the M.P.) cylinder of a three-stage expansion engine, but the arrangement is similar for every cylinder; and the whole engine with its three or more cylinders has a common seating, with sole plates joined together so as to form one complete framing.

For the reception of the engines, the ship's framing which is to support them is specially built up on the inner bottom if a double bottom is fitted, or on the inner frames if the ship is not so fitted. Double bottoms are fitted in all important ships under the machinery and boiler spaces at least, and generally in naval vessels of over 2500 tons displacement.

[FIG. 127

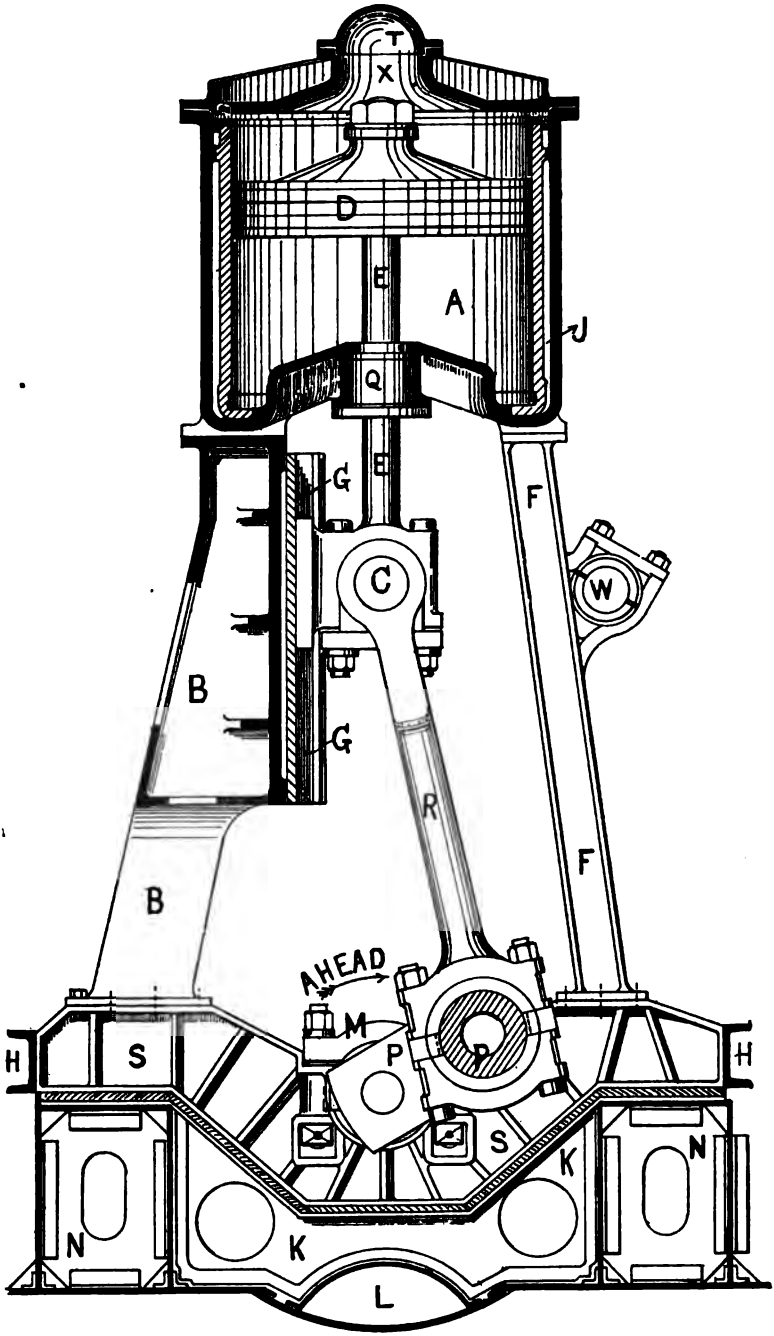


FIG. 127.—Parts of a Vertical Engine.

REFERENCES TO FIG. 127

A, Cylinder.
B, Cast-iron back frame or column.
C, Gudgeon pin at top end of connecting rod; and slipper.
D, Piston.
E, Piston rod.
F, Front columns, cast iron or forged steel.
G, Guide, or shoe; for crosshead, or gudgeon, slipper.
H, Steel beams, joining up the sole plates *S* longitudinally, and in small engines made in one piece with the sole plates.
J, Jacket steam space.
K, Engine seating, made of steel plates and frames riveted to inner bottom or transverse framing of the ship.
L, Limber hole for draining crank pits to bilge suction.
M, Main bearing for crank shaft, fitted in recess in sole plate.
N, Engine seating, made of steel plates and frames in box-shape, and connecting up the frames *K* longitudinally.
P, Crank and bottom end of connecting rod.
Q, Packing gland.
R, Connecting rod.
S, Sole plate, steel casting or forging, forms recess for main bearing.
T, Manhole cover.
W, Bearing; for weigh shaft used for reversing and controlling the link motion.
X, Cylinder cover.

Engine seating is the name generally given to the framework *K* and *N*, which is built in by the shipbuilders and riveted to the ship. The transverse frames *K* are rigidly connected, by riveting, with the box frames *N*, running longitudinally the whole length of the crank shaft. It is good practice to make the thrust-block seating continuous with the engine seating; this distributes the thrust over a considerable number of frames of the ship. The engine seating should be unyielding in shape at all times, so that the shafting should not be subject to varying alignment in its bearings, which are supported by the engine seating.

The *sole plate or bed plate* is fitted above the seating. It is a complete structure in itself, made up of the sole plate (*S* in figure) and the beams *H*, which join up the ends of the sole plates longi-

tudinally. Proper intervals and spaces must be left for the working of the cranks and eccentrics in both the sole plating and in the engine seating (see also Fig. 154). If possible, the sole plate is made of forged steel; but generally it is too complicated, and a casting is substituted. The plates are *bolted* down to the engine seating, and solid liners of cast iron are fitted (shown in cross section in the figure) between the sole plate and seating; the liners are trimmed as necessary to produce correct alignment of the bearings, in which the shafting rotates. The beams *H* are bolted to *S*, but in some cases rivets are used. Recesses are formed in the sole plates for the reception of the main bearings. Every precaution is taken to secure a rigid structure as a support for the engine *as a whole*, so that when subject to any stress it shall move as one piece and bring the least possible stress on any individual part. Non-rigidity of the framing supporting the shafting produces a varying alignment, and has frequently resulted in heated bearings and fracture of the shaft and its connections.

Nearly upright standards or *columns*, *F* and *B*, are fitted above the sole plates, and bolted down to them; nuts and check nuts are fitted on the securing bolts, and in addition, the ends of the bolts are generally riveted over to prevent any possibility of the nuts slacking back. All bolt fastenings used about the framing and columns are usually secured in the same way.

In some instances, and particularly for the ordinary cargo steamer, the front standards *F* are made in one piece, and identical with the back column *B*. Twin gudgeon pins are frequently fitted with this arrangement. *Double guides*, as above described, are particularly useful when the engines are much used for astern working, such as river steamers and tugs, because the areas of the rubbing surfaces of the slipper and guide are equal for both directions of rotation of the screw. Naval engines are not worked astern for long periods, the extra astern guide surface is not required, and double guides are seldom fitted. In addition, there is some increase in weight, and the working parts are perhaps less accessible than with the *open-fronted* framework, as shown in Figs. 124 and 127.

Light Machinery, for Torpedo Boats and Destroyers.—The machinery of torpedo boats, destroyers, and other small, speedy craft has generally pioneered the designs for larger vessels. Weight and space are of supreme importance, and it is only by the most

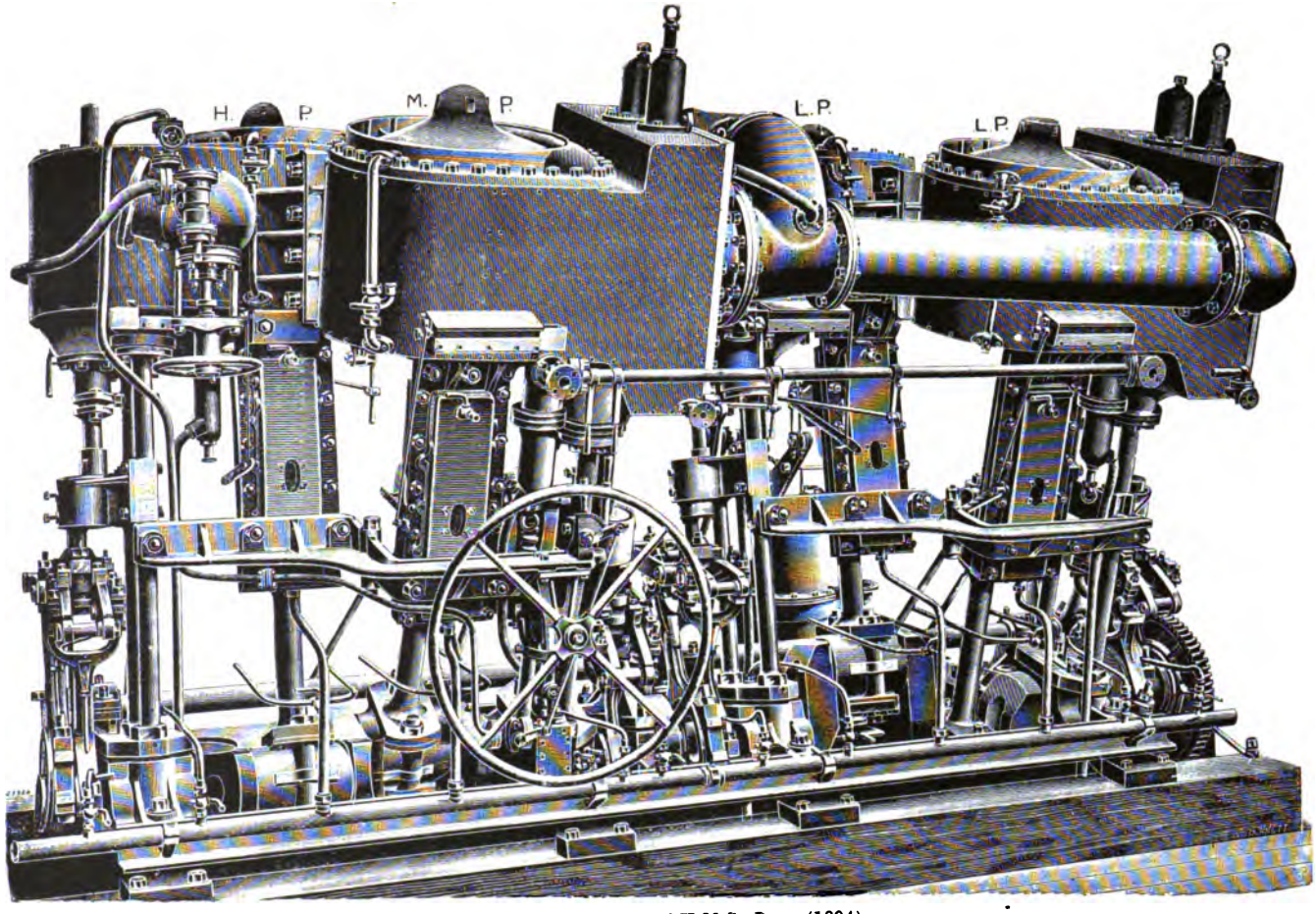


FIG. 128.—One Engine of H.M.S. *Decoy* (1894).

minute attention to the smallest details in the design of the machinery and boilers that their enormous powers have been obtained within the tonnage displacement. Fig. 128 is typical of the engines fitted in several destroyers built for a foreign power (1907). In the British Navy, all recent torpedo craft are fitted with turbine engines.

The principal difference in large and small engines is in the cylinder supports, and some saving in weight is effected by using open framework with circular pillars instead of the heavy cast-iron back frame. The cylinder supports of several United States ships are made of circular columns only.

Chocks and Ties.—The cylinders of large engines are generally secured together by tie rods, or parts of them are flanged and bolted together. The system, generally advocated for large engines, of fitting each cylinder and its supports as a separate structure, appears to be going out of fashion; this separation allowed for difference of expansion of the cylinders at top and the shaft bearings below, and thus tended to produce good alignment at all times of the rods and cranks. Although the engines are larger, and the working temperatures higher, no harm appears to have been caused by rigidly connecting the cylinders together. A good example of this is shown in Fig. 131, where the three cylinders, side by side, are bolted together. Many four-cylinder engines of the usual naval type are secured in pairs by flange joints, and each pair is secured to the other by tie rods between the joints, as shown in Figs. 124, 219, and 220.

All naval engines are fitted with ramming chocks, to relieve the working parts of the engine of the shock caused by collision or ramming, and thus prevent fracture of the machinery, in addition to the ordinary tie rods. These are not rigid connections, but arranged to come into play only when the engine is slightly displaced. When the usual side-by-side engines are fitted, the upper parts of the cylinders are sometimes stayed to the mid-line bulkhead between the two engine rooms. This arrangement adds considerably to the rigidity of the structure when the ship is heeled or rolling.

Height of Engine.—The height of an engine, which is measured from the top of the cylinder cover to the centre of the crank shaft, depends practically on the length of stroke adopted. For general purposes the connecting rod is made twice the length of stroke, and the distance between the centres of the crank and shaft is evidently

always equal to half the stroke. Putting the crank on the top centre, the height of engine = stroke + length of connecting rod + throw of crank + thickness of piston and cylinder clearance; top and bottom + height of cylinder cover + length of stuffing box and clearance allowed between it and centre of crosshead

$$= 1 + 2 + \tfrac{1}{2} + \tfrac{3}{4} + \tfrac{1}{2} + \tfrac{3}{4}, \text{ in strokes;}$$
$$= 5\tfrac{1}{2} \text{ strokes, more or less.}$$

For a destroyer of $1\frac{1}{2}$ feet stroke, the height $= 1\frac{1}{2} \times 5\frac{1}{2} = 8'\ 4''$ about.
„ a battleship of 4 „ „ $= 4 \times 5\frac{1}{2} = 22$ feet.
„ *Caronia* of 5′ 6″ stroke „ $= 5\frac{1}{2} \times 5\frac{1}{2} = 30$ feet.

(The length of connecting rod is rather over twice the length of stroke in this instance.)

Height, Space, and Weight compared.—In the formula for calculating the I.H.P. developed in any cylinder whose piston area is A, it is evident that an increase of I.H.P. can be obtained by increasing P, the mean pressure. P may be increased in two ways, either by increasing the boiler pressure and retaining the same cut-off and ratio of expansion, or by retaining the same boiler pressure and using a later cut-off. Both these methods are adopted in naval practice to save space directly and weight indirectly; the highest pressures (up to 270 lb. per square inch) at the engines and a late cut-off (generally at $\frac{3}{4}$ stroke at full power) being usual.

If P and A are limited, an increased power can be obtained by increasing the piston speed (= 2 L.N.). In the Table of Particulars of Engines it should be noticed that there has been a gradual increase in this direction, until the piston speed for battleships is about 960 feet per minute, and in destroyers and small craft it is about 1200 feet. The same piston speed can be kept by increasing N and decreasing L, while still retaining the same I.H.P. This decrease of stroke L reduces the height of the engine and, consequently, the space occupied by it, and also its weight. The increased number of revolutions decreases the size of shafting necessary (see Chapter XV. for formula for diameter of shaft), and also of its connections; this also contributes to reduced weight and space. The above conclusions explain why the same I.H.P. is obtainable with considerably less weight and space in a destroyer whose engines move at a rate of 400 revolutions per minute than in a battleship whose engines, of 4 feet stroke, make 120 revolutions per minute.

For the same reasons the engines of a large mail steamer, say the *Caronia*, with 5′ 6″ stroke and 100 revolutions per minute, are heavier and occupy more space than those of a battleship, although the factor of safety, or margin of strength, allowed may be the same.

Higher pressures involve stronger engines, with stronger attachments and working parts, so that little economy in weight is effected directly from their use. But indirectly, an economy of fuel is obtained by using a greater rate of expansion with the higher pressure, so that a somewhat smaller engine is necessary for the same I.H.P.

Economy of weight and space are effected in modern machinery by—

1. Increased pressures, producing greater mean pressures on the pistons.
2. Increased speed of piston and revolution.
3. Better materials, allowing greater working stress for the weight employed.
4. Using greater rates of combustion in the boiler furnaces.
5. By using water-tube boilers, particularly those with small (upright) tubes.

Number and Arrangement of Cylinders.—The number of cylinders has been gradually increased, and, where turbines have not been adopted, the tendency is still in this direction. Very large cylinders require much staying, and the piston rod must also be proportionately large to withstand the possible bending stresses which are put on it. Although a rod may be quite *strong* enough, it must be also made *stiff* enough for its work; this remark is also true particularly of large pistons. In the merchant service, where height is not of so much consequence, the increase in number of cylinders does not necessarily increase the number of cranks on the shafting, as the cylinders can be arranged tandem, one above the other, with a piston rod common to two cylinders. This arrangement has been followed in many cases where alterations have been made to introduce a greater number of stages of expansion to engines already fitted in ships. The present usual arrangement is to fit the cylinders *side by side*, each cylinder having its connecting rod attached to its own crank. This arrangement is always fitted in warships, to keep the height as low as possible; but for some auxiliary engines, which are adapted for compound working, the tandem arrangement is now used.

Four-cylinder, Three-stage Expansion Engines.—For naval service in recent designs, the large L.P. cylinder has given place to two smaller ones of similar total capacity. The two L.P. cylinders make the engine rather longer, the frictional loss is greater with four cranks and connections than with three, and there is also a slight increase in weight; these are the disadvantages. The advantages which, however, overcome them are:—

1. Better balancing of the engines as a whole, producing freedom from vibration with the very fast-running engines as fitted in destroyers and small craft, and generally in all modern warships.

2. The great increase in the size of L.P. cylinders required for large powers in the bigger battleships and cruisers makes it extremely difficult to construct cylinders and pistons for safe working; the exceedingly rapid reciprocation of the piston enormously increases its momentum, and there is consequently less difficulty in producing smaller pistons of sufficient stiffness and strength.

Each of the two L.P. cylinders is made of one-half the volume of the single cylinder otherwise required, but each, with its connections, is made strong enough to develop about five-eighths of the power developed in each of the H.P. and M.P. cylinders. This increase of strength is particularly convenient with the closed exhaust system, where some of the steam from the auxiliary exhaust system is made to do work in the L.P. cylinders before passing into the condenser. A plan of the recent naval arrangement of cylinders, etc., is shown in Fig. 129.

Four-stage Expansion.—The naval engine is a compromise which embraces the greatest possible power on the least possible weight, in the least possible space, with a moderate economy at full power, and a relatively low fuel consumption at cruising speeds. The adoption of four-stage expansion would probably result in a large increase of fuel consumption at cruising speeds, an increase in weight and space, and a consequent great increase of tonnage and first cost. As an example of the increased weight of machinery involved in mercantile practice, that of the ss. *Saxonia* is actually double that of H.M.S. *Minerva* or *Hyacinth*, and the power is slightly less, although possibly more easily maintained.

Ships of the mercantile marine are built to steam at some

particular speed, which is obtained by exerting the engines at very nearly their full power; economy of fuel, and not economy

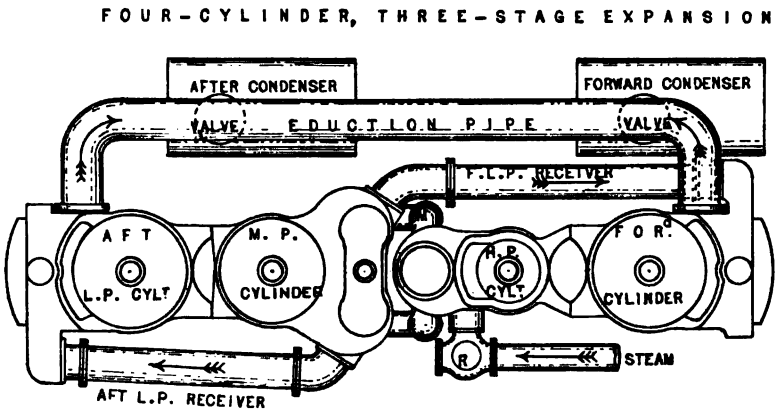


FIG. 129.—Four-cylinder, Three-stage Expansion Engine.

of weight and space, is the primary object. The extra weight and space under these conditions is soon saved in economy of fuel.

The twin engines of the Orient-Pacific ss. *Orontes* are a good example of mercantile practice with four-stage expansion. The working pressure is 215 lb. per square inch, and the other details are shown in the tabulated statement at the end of the

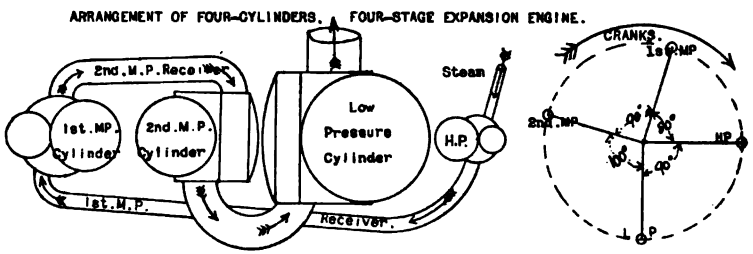


FIG. 130.—Four-stage Expansion Engine.

chapter. The engines are of the usual side-by-side arrangement. The H.P. cylinder is placed forward, next abaft is the L.P., then the 2nd M.P., and abaft all the 1st M.P. cylinder. The engines are balanced on the principle of varying crank angles, obtained from the calculation of inertia forces and couples; the H.P. crank leads the 1st M.P. crank by about 80°, followed by the 2nd M.P. at about 90°, which is about 100° in advance of the L.P. crank,

leaving about 90° between the last and first. The crank angles and course of the steam are shown in Fig. 130.

"Kaiser Wilhelm II."—The general arrangement of one of the eight-cylinder engines of this twin-screw steamship is shown in Fig. 131. There are two engine rooms to each shaft, and four

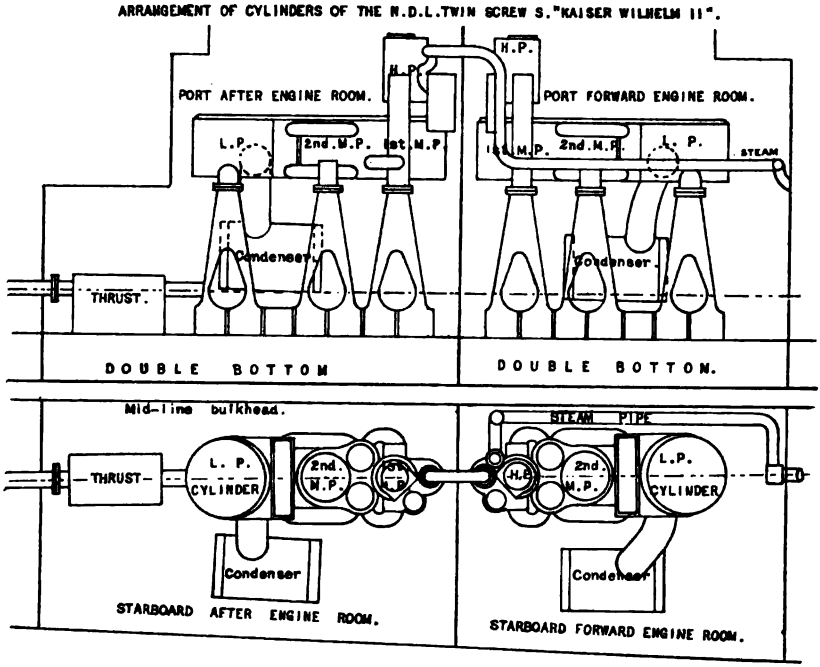


FIG. 131.—Engines of ss. *Kaiser Wilhelm II.*

cylinders in each. The H.P. and 1st M.P. cylinders are placed tandem, the H.P. being the upper; and the 2nd M.P. and L.P. cylinders are abaft them in the after engine room, and before them in the forward engine room. The dimensions are given at the end of the chapter. These engines are notable for the great power which is transmitted by each shaft, a power which a few years ago was considered too great for any single shaft to carry. The general details of internal construction follow the usual designs of this country, and the proportions are similar.

SS. "Caronia" (Cunard Line).—Outlines of the engines—four-cylinder, four-stage expansion—of the ss. *Caronia* are shown in Fig. 132. The *Caronia* is a sister vessel to the *Carmania*, fitted

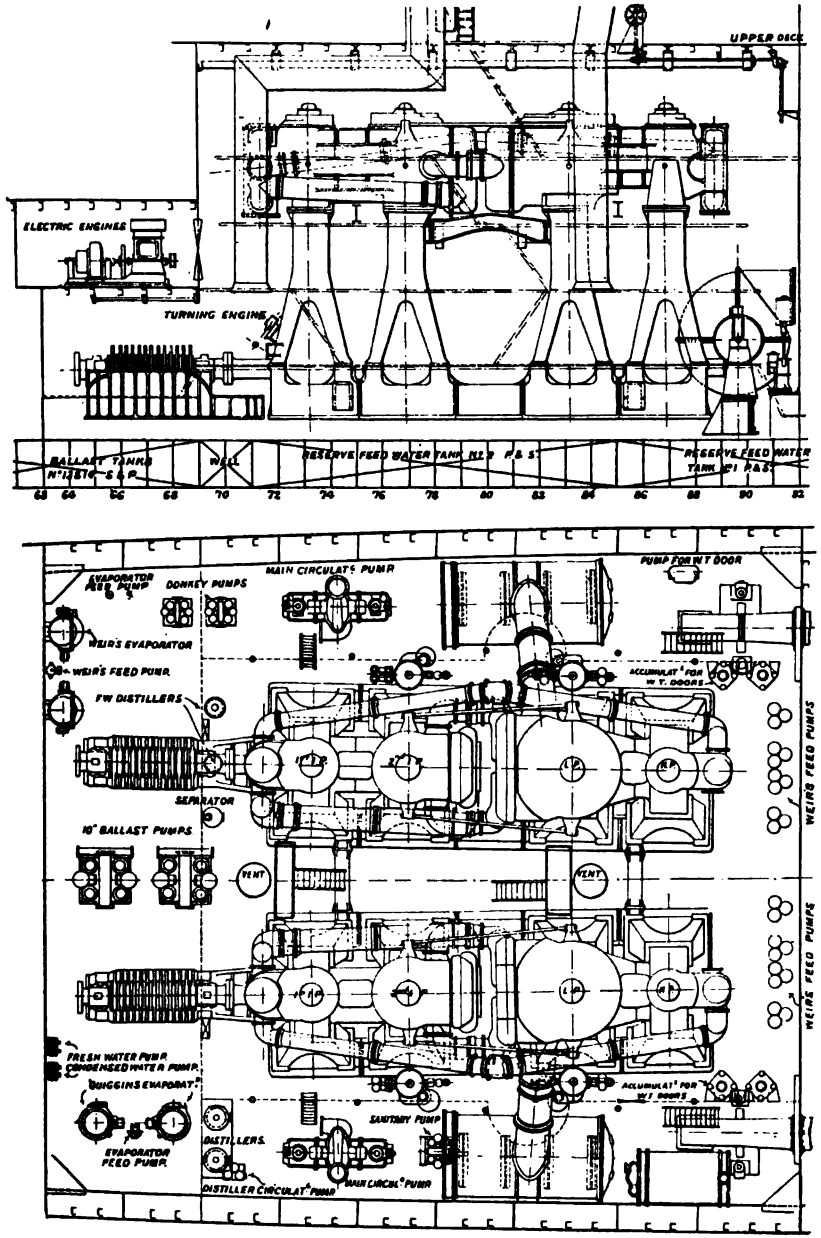


FIG. 132.—Engines of ss. *Carinia*.

with turbines. Some particulars are given in the table, but others of interest are:—

Length of connecting rods	12 feet.
Total height from centre of shaft . . .	30 „
„ „ bottom of crank-pits . .	36 „
Height of funnel	135 „

Howden's system of hot-air draught.

2·35 square feet of boiler heating surface is allowed per I.H.P., and about 17½ lb. of coal are burnt per square foot of grate area per hour, and thus the draught created by the funnel is almost, if not entirely, sufficient, without mechanical draught or other assistance. Coal per I.H.P. per hour (on trial) was 1·29 lb. and the steam consumption 13·4 lb. per I.H.P. per hour, showing that 11·3 lb. of steam were produced for 1 lb. of coal burnt, which gives a very high boiler efficiency.

Balancing of Engines.—When an unbalanced engine rotates, there are certain forces transmitted to the bed-plate through the crank-shaft. These forces are, roughly:—1st. The centrifugal forces set up by masses, such as the crank-arms, etc., rotating round the centre line of the crank-shaft and continuously changing direction. 2nd. Forces set up by masses, such as the crossheads, etc., reciprocating in a straight line. If not counteracted or balanced, these forces will cause vibration. One method is to fit balance-weights on the crank-shaft to set up equal and opposite forces to the centrifugal forces, and bob-weights to set up equal and opposite forces to those caused by the reciprocating parts. In actual calculations a portion of the mass of the connecting rod is taken as a rotating, and the remainder as a reciprocating, mass.

The above method is obviously cumbersome, and modern marine engines are generally balanced on the *Yarrow-Schlick-Tweedy system*, which is effected by arranging the relative positions of the cylinders according to the size and weight of their moving parts, and by making the angles between the various crank radii such that the forces set up by the moving parts connected with the various cylinders counteract one another and so leave no unbalanced forces.

It is not within the scope of this book to enter into the abstruse calculations and higher mathematics necessarily involved in the correct balancing of engines.

Figs. 129 and 132 may be considered as typical of modern naval and mercantile practice in balancing engines. The lighter reciprocating (connecting rod) masses are placed near opposite ends of the crank-shaft, while the heavier ones are near its mid-length. The centre lines of the cranks (and cylinders) are closely pitched for each pair near the ends, leaving as great a space as practicable between the two middle cranks. In the naval engine the L.P. connecting rods and connections, having less power to transmit, are lighter than the other two; in the mercantile engine the H.P. and 1st M.P. pistons are lighter, being smaller than the other two. The slide valves are so arranged that the pairs of cranks are brought closely together in each case; but the heavier L.P. slide valves are placed near the ends in the naval engine, while the opposite is the case with the mercantile engine. These positions are chosen in conjunction with the more important position of the cranks. The calculation to be complete must take into account the slide gear as well as the connecting rod masses, the effect of obliquity of both connecting and eccentric rods, and the proper timing of the operations of the slide valve, so as to neutralise as far as possible its varying inertia. A matter of first importance in relation to the balancing of engines is the rigidity of the engine framing in conjunction with the bracing of the cylinders.

Particulars of Engines.—A table is given below of a few particulars of some typical engines. It will be seen that there has been a steady increase of piston speed, and a disposition to adopt a shorter stroke for naval engines, with a consequent increase of speed of revolution. The curtailment of the length of the stroke, however, appears to have been reached for the various types of ship. The piston speed for destroyers and other small craft has been increased up to about 1200 feet per minute, and, as experience is gained, it is probable that if turbines had not been adopted the speed would have been increased gradually in the larger vessels. Increased piston speed means a lighter engine, decreased stroke means less height of engine—both desirable from a naval point of view if they can be used with safety

[TABLE

Some Particulars of Engines

Battleships

Date.	I.H.P.	Boilers.	Number.	Pressure.		Cylinders, Diameters.				Ratio L.P. to H.P.	Stroke.	Revolutions.	Piston Speed.	Ship Speed.
				Boilers.	Engines.									
						in.	in.	in.	in.		ft. in.			
1895	12,000	Cylindrical	8	155	150	40	59	88		4·84	4 3	100	850	17·5
1897	13,500	Belleville (Economiser)	20	300	250	30	49	80		7·1	4 3	108	918	18·25
1898	15,000	,, ,,	20	300	250	31½	51½	84		7·1	4 3	108	918	18·0
1901	18,000	,, ,,	24	300	250	33½	54½	63	63	7·1	4 0	120	960	19·0
1903	18,000	Babcock	...	270	250	33½	54½	63	63	7·1	4 0	120	960	18·50
1904	18,000	W.T. and cyl. (3)	...	210	205	38	60	70	70	6·28	4 0	120	960	19·0
1905	16,500	W. T. Yarrow	18	270	250	32¾	52¾	60	60	6·7	4 0	120	960	18·5
						First-class Cruisers								
1896	16,500	Belleville (Economiser)	30	300	250	34	55½	64	64	7·1	4 0	110	880	20·5
1898	18,000	,, ,,	30	300	250	34	55½	64	64	7·1	4 0	120	960	20·75
1900	21,000	,, ,,	30	300	250	36	59	68	68	7·1	4 0	120	960	21·0
1901	22,000	,, ,,	31	300	250	37	60	69	69	7·1	3 6	140	980	23·0
1901	30,000	,, ,,	43	300	250	43½	71	81½	81½	7·0	4 0	120	960	24·0
1904	21,000	W.T. (17) and cyl. (6)	...	210	205	41½	65½	73½	73½	6·28	3 6	138	966	22·25
						Second-class Cruisers								
1898	10,000	Belleville (Economiser)	18	300	250	26	42	48	48	6·84	2 6	180	900	20·0
1903	12,500	Dürr	16	270	250	26	42	48	48	6·8	2 8	180	960	22·0
						Mercantile Marine								
1900 (*Saxonia*)	9,000	Cylindrical	9	210	210	29	41½	59	84	8·36	4 6	80	720	16·0
1903 (*Orontes*)	10,000	Cyl. (4 S.E. + 2 D.E.)	6	215	210	27¼	39	56	80	8·62	6 0	...	...	18·25
1906 (*Caronia*)	20,000	Cyl. (5 S.E. + 13 D.E.)	...	210	200	39	54½	77	110	7·95	5 6	82	900	19·00
						"Kaiser Wilhelm II."								
1903	45,000	Cylindrical	...	215	210	37½	49¼	74¾	112¼	9·0	5 11	80	945	23·5

Dimensions, etc., of Notable Atlantic Steamers

Steamer's Name.	Builders.	Date.	Moulded Dimensions.			Proportion of Length.		Draught.	Displacement.	Gross Tonnage.	Cylinders.		Boilers.			Indicated Horse-Power.	Speed on Trial.
			Length.	Breadth.	Depth.	To Beam.	To Depth.				Diameter in Inches.	Stroke.	Heating Surface.	Grate Area.	Working Pressure.		
			ft.	ft. in.	ft. in.			ft. in.	tons.			in.	sq. ft.	sq. ft.	lb.		knots.
Great Eastern	Scott Russell	1858	680	83 0	57 6	8·192	11·826	25 6	27,000	24,360	Screw, four 84-in.; paddle, four 74-in.	S., 48-; P., 17-	..	..	30	7,650	14·5
Britannic	Harland & Wolff	1874	455	45 0	36 0	10·111	12·640	23 6	8,500	5,004	Two 48-in., two 83-in.	60	..	..	70	5,500	16
Arizona	Fairfield Co.	1879	450	45 2	37 6	9·955	12·000	22 0	..	5,147	One 62-in., two 90-in.	66	..	..	90	6,300	17
Servia	Clydebank Works	1881	515	52 0	40 6	9·903	12·716	23 3	9,900	7,392	One 72-in., two 100-in.	78	27,483	1014	90	10,300	17
Alaska	Fairfield Co.	1881	500	50 0	39 8	10·00	12·607	22 0	..	6,932	One 68-in., two 100-in.	72	..	..	100	10,500	18
City of Rome	Barrow Co.	1881	543	52 0	38 9	10·432	14·000	22 0	11,230	8,141	Three 46-in., three 86-in.	72	29,286	1398	90	11,900	18·23
Aurania	Clydebank Works	1882	470	57 0	39 0	8·245	12·051	20 9	8,800	7,269	One 68-in., two 91-in.	72	23,284	1001	90	8,500	17·5
Oregon	Fairfield Co.	1883	500	54 0	40 0	9·259	12·500	23 0	10,500	7,375	One 70-in., two 104-in.	72	38,047	1428	110	13,300	18·3
America	Clydebank Works	1884	432	51 0	38 0	8·470	11·520	23 0	9,300	6,500	One 63-in., two 91-in.	66	22,750	882	95	7,354	17·8
Umbria	Fairfield Co.	1884	500	57 0	40 0	8·772	12·500	22 6	10,500	7,718	One 71-in., two 105-in.	72	38,817	1606	110	14,321	20·18
Lahn	,,	1887	448	48 10	36 6	9·174	12·247	23 0	7,700	5,661	Two 32½-in., one 68-in., and two 85-in.	72	..	..	150	8,900	17·78
Paris	Clydebank Works	1888	528	63 0	41 10	8·373	12·610	23 0	13,000	10,499	Two 45-in., two 71-in., and two 113-in.	60	50,265	1293	150	20,000	21·8
Augusta Victoria	Vulcan Company, Stettin.	1889	460	55 6	39 0	8·288	11·795	22 9	9,500	7,661	Two 41⅞-in., two 66⅛-in., and two 106⅞-in.	63	36,000	1120	150	14,110	18·31
Columbia	Laird Brothers	1889	463	55 6	39 0	8·333	11·860	22 9	9,500	7,578	Two 41-in., two 66-in., and two 101-in.	66	34,916	1226	150	13,680	19·15
Teutonic	Harland & Wolff	1890	565	57 6	42 2	9·826	13·425	22 0	12,000	9,686	Two 43-in., two 68-in., and two 110-in.	60	40,072	1154	180	19,500	21
Normannia	Fairfield Co.	1890	500	57 3	38 0	8·730	13·150	22 0	10,500	8,716	Two 40-in., two 67-in., and two 106-in.	66	46,490	1452	160	16,352	20·78
Spree	Vulcan Company, Stettin.	1890	463	51 6	37 6	9·00	12·346	22 0	8,900	6,963	Two 38-in., one 75-in., and two 100-in.	72	..	..	165	13,000	19·6
Fürst Bismarck	Vulcan Company, Stettin.	1891	503	57 3	38 0	8·777	13·224	22 6	10,200	8,000	Two 43⅜-in., two 67-in., and two 106¼-in.	63	47,000	1450	157	16,412	20·7
Campania	Fairfield Co.	1893	600	65 0	41 6	9·231	14·457	23 0	18,000	12,500	Four 37-in., two 79-in., and four 98-in.	60	82,000	2630	165	30,000	22·01
St. Louis	Cramp, Philadelphia	1895	536	63 0	42 0	8·50	12·75	26 0	16,000	11,629	Four 28-in., two 55-in., two 77-in., four 77-in.	60	40,320	1144	200	18,000	21·08
Kaiser Wilhelm der Grosse.	Vulcan Company, Stettin.	1897	625	66 0	43 0	9·46	14·27	28 0	20,880	14,349	Two 52-in., two 89-in., and four 96·4-in.	68·8	84,285	2618	178	30,000	22·5 to 23
Oceanic	Harland & Wolff	1899	685	68 5	49 0	10·01	13·98	32 6	28,500	17,274	Two 47·5-in., two 79-in., and four 93-in.	72	74,686	1962	192	27,000	20·72
Deutschland	Vulcan Company, Stettin.	1900	662·9	67 0	44 0	9·89	15·06	29 0	23,620	16,502	Four 36·61-in., two 73·6-in., two 103·9-in., and four 106·3-in.	72·8	85,468	2188	220	36,000	23·25 to 23·5
Kronprinz Wilhelm.	Vulcan Company, Stettin.	1901	663	66 0	43 0	..	..	29 0	21,300	15,000	Four 34·2-in., two 68·8-in., two 98·4-in., and four 102·3-in.	70·8	93,685	2702	213	36,000	23·25 to 23·5
Kaiser Wilhelm II.	Vulcan Company, Stettin.	1903	678	72 0	52 6	9·41	12·9	29 0	26,000	20,000	Four 37·4-in., four 49·2-in., four 74·8 in., and four 112·2-in.	70·86	107,643	3121	225	38,000	23·5
La Provence	St. Nazaire Works	1906	597 ft. 1½ in.	64 7½	41 8	9·23	14·35	26 9	19,160	13,750	Two 47·2-in., two 76·2-in., four 88·18-in.	66·9	58,342	1571	200	30,000	22·05
Lusitania	Clydebank Works	1907	760	88 0	60 4½	8·55	12·56	33 6	38,000	32,500	Turbines	..	158,850	4048	195	68,000	25
Mauretania	Swan, Hunter, and Wigham-Richardson	1907	790	88 0	60 6	..	..	..	38,000	32,000	Turbines	..	159,000	4060	195	68,000	25

WEIGHTS AND PARTICULARS OF BOILERS AND ENGINES

	Six *Duncans.*	Two *Triumphs.*	*Minotaur.*	*Shannon.*
Total machinery and boiler weights	1600	1000	2430	2410
Boiler weights	745	440	...	...
I.H.P. from Navy List	18,000	12,500	27,000	27,000
I.H.P. per 1 ton weight (total)	11·25	12·5	11·1	11·2
,, ,, (boilers only)	14·1	28·3	...	Saves 20 tons on Babcock
,, ,, (engines only)	21·0	22·3	...	
Steam pressure, boilers	300	275	275	275
,, engines	250	250	250	250
Boilers	Belleville	Yarrow	Babcock	Yarrow
Number	24	12	25	24
Grate area, sq. ft.	1375	664	1958	1350
Heating surface, sq. ft.	43,760	37,524	73,000	81,000
I.H.P. per 1 sq. ft. of grate area	13·1	18·8	13·8	20·0
Heating surface per 1 I.H.P.	2·43	3·0	2·7	3·0
Ratio of heating surface to grate	32	56·5	35·5	60·0
Engines—				
Cylinder diameters, H.P. and M.P.	33½, 54½	29, 47	40⅝, 65½	40⅝, 65½
,, ,, for 2 L.P.	63, 63	54, 54	74⅝, 74⅝	74⅝, 74⅝
Stroke	48	39	48	48
Revolutions per minute at full power	120	150	129·5	127
Piston speed, feet per minute	960	975	1036	1016
Ship's speed knots	18·92	20·0	23·0	22·4
Coal per 1 I.H.P. per hour	2·02 lb.	1·73 lb.	1·59 lb.	2·01 lb.
Displacement tonnage	14,000	11,800	14,600	14,600

CHAPTER XIV

CYLINDERS AND THEIR FITTINGS

Cylinders.—The cylinder is a very complicated casting, and it is almost impossible to make it satisfactorily in steel, so that cast iron is generally employed, stayed as necessary to give it local strength.

When the piston is descending, the cylinder has to withstand the resistance of the upward pressure opposed to the piston; this means that the cylinder must be supported in such a way that it cannot yield to this pressure, and must be securely held down. In a similar way it must be capable of withstanding the downward pressure when the piston is rising; consequently the method of securing it down to the supports on which it rests must be proportioned to these stresses. In Fig. 133 the cylinder feet are clearly shown; the rear foot is generally secured to the back frame, or column, which also supports the guide, and the front supports are generally cylindrical columns or pillars.

The bottom or front end of the cylinder generally forms part of the casting, but covers are fitted at the back end so that the piston and slide valve, and, if required, the liners, working barrels, and false faces, can be admitted.

The H.P. cylinder and slide casing are generally made in one piece of hard close-grained cast iron, with feet, as shown in Fig. 133.

The M.P. cylinder and slide casing follow the same course, where a cylindrical valve is employed, as is now usual; but if a flat valve is fitted, then the casting follows the practice for the L.P. cylinder and casing, which is made as shown in Fig. 134.

Cylinder False Face.—The working face of the cylinder is lined where the slide valve moves on it, in a manner somewhat similar to that provided for the piston.

When cylindrical valves are employed, as is now usual for the

H.P. and M.P. slide valves, the face is made in two parts, as shown in Fig. 133. The inner, or bottom, face is a cylindrical barrel, fitted with the necessary steam ports and an inward flange, which is jointed to the cylinder casting in the same way as the lower end of the cylinder liner. This inner face must be of slightly smaller diameter than the outer one, so that it may be passed into its

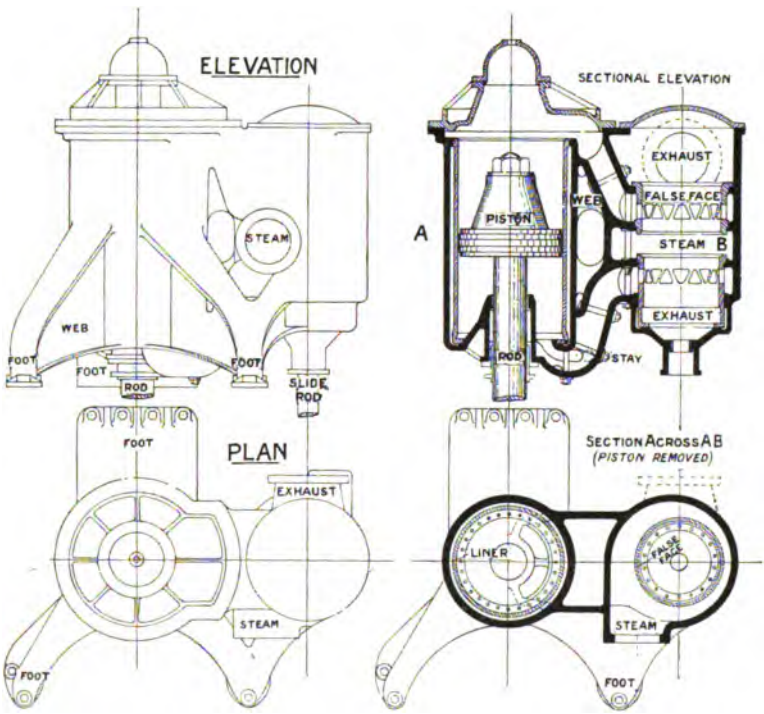


FIG. 133.—H.P. Cylinder.

place. The outer, or upper, face is made with an outward flange at its top end; this is necessary for getting the valve into place, and also for its subsequent working. The joints are each made with red-lead in the usual way, and the bolts are secured by either guard rings and studs, or the bolt heads are caulked at one corner. Separate parts are preferred for the upper and lower faces, because, if made in one piece, there is some difficulty from unequal expansion in making both joints steam-tight. The H.P. and M.P. false faces are made of hard close-grained cast iron.

For flat valves, the face is made in one piece (Fig. 134), and secured to the cylinder casting by cheese-headed screws. The same material is used as for the other false faces.

Steel Stays, as shown in Fig. 133, are used for the flat surfaces subject to pressure in the cylinder ports (or steam ports, as they are more usually called). These stays are fitted with nuts on the outside, and bosses are made on the casting for bringing up the strength locally—to at least an equivalent to the stays employed.

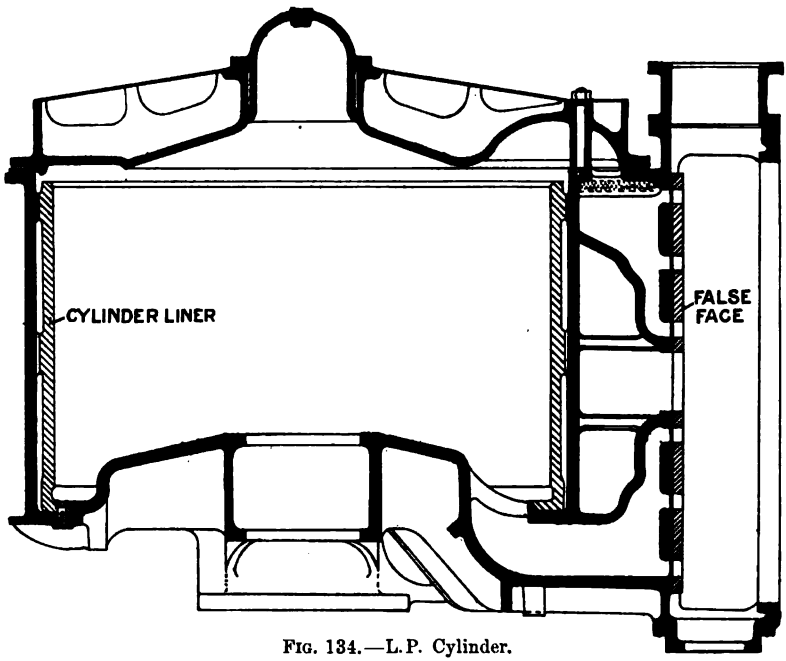


FIG. 134.—L.P. Cylinder.

In some cases the stay is fitted into a long-sleeved hole, so that it is not in actual contact with the steam. The *steel* stay and the *cast-iron* sleeve expand unequally, and stresses are thus brought on the casting of the cylinder; therefore the naked stay is preferable, although a steam joint has to be made at its outer end. The nuts are generally riveted over on the end of the stay, to prevent their slacking back.

In addition to the stays, *webs* are made wherever possible in the casting, to give local strength to it; but care must be taken that this webbing and staying are consistent with the expansion of

the cylinder itself, and that they shall not induce fracture by their unequal expansion when the engine is warmed up to its work; many fractures have occurred from this cause alone.

The cylinder **covers** are made of cast steel. In a few cases, however, they have been made of forged or stamped steel; but this is only possible when the form is simple, and without any complications of webbing. The interior of the cover follows the shape of the piston as closely as possible; this reduces the clearance spaces. The exteriors are made with webs of T section for those fitted radially from the manhole flange (Figs. 133 and 134). A vertical web, sometimes of inverted L section, is made far enough from the edge to leave the heads of the nuts clear of it; this follows the shape of the edge of the cover.

Lagging.—All projections on the cylinder casting are made so that it is not necessary to disturb the lagging when fitting or jointing up the connections, steam or eduction pipes, drains, ramming stays, indicator cocks and pipes, jacket valves and drains, etc.

The whole of the hot surfaces of the cylinders, slide-valve casings and covers, cylinder covers and pipes, are covered with asbestos or other bad conductor of heat. Mats of asbestos fibre are used wherever possible, and all parts are then covered over with asbestos cloth. To keep the lagging in place it is covered up with strips of closely fitting mahogany (or sheet lead for bottoms of cylinders), bound together with brass ribbons and screws. The foundations to which the outer strips are fixed are preferably made of some incombustible material—not wood,—and a form of asbestos is generally used.

Cylinder Liners.—The working surfaces on which the piston and slide valve move are generally made separate from the cylinder casting; this reduces its complexity, and provides more suitable material for the working surfaces.

The L.P. cylinder liner, or working barrel, is made of hard close-grained cast iron (Figs. 134 and 137).

The H.P. and M.P. cylinder liners are made of forged steel; and are thus made thinner for the greater pressure which they have to stand, than if made of cast iron (Figs. 133 and 135).

The space between the cylinder and the liner is utilised as a *jacket steam-space*, generally from 1 to $1\frac{1}{4}$ inches in width. To prevent the steam in the jacket leaking into the cylinder, and *vice*

versa, it is necessary to make the ends of the cylinder liner steam-tight at the inner surface of the cylinder.

The inner end (the bottom in a vertical inverted engine) of the liner is flanged inwards, and fits accurately into a recess cut for it at the end of the cylinder, and makes a joint with it, as shown in Figs. 135 to 139. The type shown in Fig. 136 is now usually employed for all cylinders; the tops of the heads of the bolts are below the level of the flange. The jointing material is a thin red-lead mixture, with a strand or two of asbestos twine to prevent it from being blown out of its place; but where the face of the flange or cylinder is not very true, it is better to use asbestos sheeting, about $\frac{1}{16}$-inch thick. The bolts are made with square heads, and are tightened up by using a long spanner from outside the cylinder; but, if possible, a short socket spanner and tommy are more effective. To prevent the bolts slacking back, they are caulked in at a point on the collar; and if nuts are fitted on the outside, split pins are fitted, to prevent the nuts slacking.

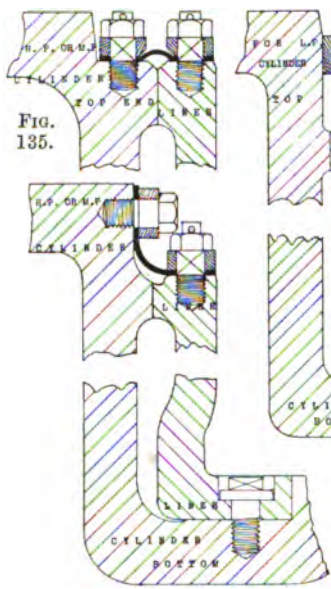


FIG. 135.

FIG. 137.

FIG. 136.

The outer end (or top, in a vertical inverted engine) of the liner cannot be jointed in the same way as the bottom end, because there is unequal expansion of the steel of the liner and of the cast iron of the cylinder to be allowed for, and therefore an expansion joint is fitted at this top end.

Various methods are shown in Figs. 135, 136, 137. In addition to making a tight joint, allowing for independent expansion of the cylinder and liner, it is preferable to make it capable of adjustment and repair without disturbing the flange joint of the liner.

For the H.P. and M.P. cylinders a flat ring, either with or without a single upward (Fig. 135) or downward corrugation, is the

best for ordinary use, and most easily jointed up; but it entails a slightly increased diameter of cylinder, at the flange, to contain it. The copper expansion ring is stiffened under the nuts by two iron or steel annular washer rings, as shown, and these are rounded off on one edge to allow movement of the expansion ring. The studs securing the rings are made with square, or pear-shaped, necks, where they pass through the supporting rings, to prevent them slacking back. In some cases, collars are made on the studs under the ring; but if the previous method is used, the collars are unnecessary, although frequently fitted. The copper angle ring, with a well-rounded corner (Fig. 136), is used where space and weight are of more importance than convenience in making the joints. The bolts and studs are in this case fitted alternately in the joint on the cylinder and the joint on top of the liner. The liner joint must be made first, and the difficulty of making and spreading the two joints so that they shall both be efficient is manifest. The studs are secured in the usual way, and the corners of the heads of the bolts are caulked, to prevent them slacking back.

For the L.P. cylinder a ring with a flange, as shown in Fig. 137, is used to press down the packing caulked into the recess below. The packing used is generally asbestos rope, well covered with black-lead, and smeared with sufficient mineral grease to make the black-lead adhere to the packing. This is the simplest form of *liner expansion joint*, and is generally sufficient for low pressures; but it is not thought good enough for the H.P. and M.P. cylinders, although experience has proved it reliable.

Pistons.—A cast-iron piston of the flat or hollow type, which was frequently fitted, is now seldom used in large cylinders. For small engines it is still very common, either with or without a junk ring, the use of which is explained later. The front and back faces of a flat piston are strengthened by radial webs between them, and for moderately high pressures and diameters one face, generally back, is coned to give increased strength.

As pressures increased, it became impossible to build flat cast-iron pistons of sufficient strength without a great increase of weight, so that now all pistons, for large engines, are made of steel, and generally forged into the shape of a cone, which is unyielding in shape under the pressure to which it is subject. This also embodies a minimum weight. The height of the cone necessary for the largest piston is first determined, and then this height is generally

Figs. 129 and 132 may be considered as typical of modern naval and mercantile practice in balancing engines. The lighter reciprocating (connecting rod) masses are placed near opposite ends of the crank-shaft, while the heavier ones are near its mid-length. The centre lines of the cranks (and cylinders) are closely pitched for each pair near the ends, leaving as great a space as practicable between the two middle cranks. In the naval engine the L.P. connecting rods and connections, having less power to transmit, are lighter than the other two; in the mercantile engine the H.P. and 1st M.P. pistons are lighter, being smaller than the other two. The slide valves are so arranged that the pairs of cranks are brought closely together in each case; but the heavier L.P. slide valves are placed near the ends in the naval engine, while the opposite is the case with the mercantile engine. These positions are chosen in conjunction with the more important position of the cranks. The calculation to be complete must take into account the slide gear as well as the connecting rod masses, the effect of obliquity of both connecting and eccentric rods, and the proper timing of the operations of the slide valve, so as to neutralise as far as possible its varying inertia. A matter of first importance in relation to the balancing of engines is the rigidity of the engine framing in conjunction with the bracing of the cylinders.

Particulars of Engines.—A table is given below of a few particulars of some typical engines. It will be seen that there has been a steady increase of piston speed, and a disposition to adopt a shorter stroke for naval engines, with a consequent increase of speed of revolution. The curtailment of the length of the stroke, however, appears to have been reached for the various types of ship. The piston speed for destroyers and other small craft has been increased up to about 1200 feet per minute, and, as experience is gained, it is probable that if turbines had not been adopted the speed would have been increased gradually in the larger vessels. Increased piston speed means a lighter engine, decreased stroke means less height of engine—both desirable from a naval point of view if they can be used with safety

[TABLE

Some Particulars of Engines

Battleships

Date.	I.H.P.	Boilers.	Number.	Pressure. Boilers.	Pressure. Engines.	Cylinders, Diameters.	Ratio L.P. to H.P.	Stroke.	Revolutions.	Piston Speed.	Ship Speed.
						in. in. in. in.		ft. in.			
1895	12,000	Cylindrical	8	155	150	40 59 88	4·84	4 3	100	850	17·5
1897	13,500	Belleville (Economiser)	20	300	250	30 49 80	7·1	4 3	108	918	18·25
1898	15,000	,, ,,	20	300	250	31½ 51½ 84	7·1	4 3	108	918	18·0
1901	18,000	,, ,,	24	300	250	33½ 54½ 63 63	7·1	4 0	120	960	19·0
1903	18,000	Babcock	...	270	250	33½ 54½ 63 63	7·1	4 0	120	960	18·50
1904	18,000	W.T. and cyl. (3)	...	210	205	38 60 70 70	6·28	4 0	120	960	19·0
1905	16,500	W. T. Yarrow	18	270	250	32¾ 52¾ 60 60	6·7	4 0	120	960	18·5
						First-class Cruisers					
1896	16,500	Belleville (Economiser)	30	300	250	34 55½ 64 64	7·1	4 0	110	880	20·5
1898	18,000	,, ,,	30	300	250	34 55½ 64 64	7·1	4 0	120	960	20·75
1900	21,000	,, ,,	30	300	250	36 59 68 68	7·1	4 0	120	960	21·0
1901	22,000	,, ,,	31	300	250	37 60 69 69	7·1	3 6	140	980	23·0
1901	30,000	,, ,,	43	300	250	43½ 71 81½ 81½	7·0	4 0	120	960	24·0
1904	21,000	W.T. (17) and cyl. (6)	...	210	205	41½ 65½ 73½ 73½	6·28	3 6	138	966	22·25
						Second-class Cruisers					
1898	10,000	Belleville (Economiser)	18	300	250	26 42 48 48	6·84	2 6	180	900	20·0
1903	12,500	Dürr	16	270	250	26 42 48 48	6·8	2 8	180	960	22·0
						Mercantile Marine					
1900 (*Saxonia*)	9,000	Cylindrical	9	210	210	29 41½ 59 84	8·36	4 6	80	720	16·0
1903 (*Orontes*)	10,000	Cyl. (4 S.E. + 2 D.E.)	6	215	210	27¼ 39 56 80	8·62	6 0	...	...	18·25
1906 (*Caronia*)	20,000	Cyl. (5 S.E. + 13 D.E.)	...	210	200	39 54½ 77 110	7·95	5 6	82	900	19·00
						"Kaiser Wilhelm II."					
1903	45,000	Cylindrical	...	215	210	37½ 49¼ 74¾ 112¼	9·0	5 11	80	945	23·5

end of the rod *R*; by screwing the eyebolt further in, the piston rod is forced back and freed from the piston *D*. For the piston shown in Fig. 138, a flange is made at the top of the piston; under this flange two half-clips are bolted together, and then two long bolts with a strongback are used to support a central screw. For small pistons two or more holes are drilled in the upper end of the piston, covered by the nut when in place; studs fit into these holes, and with a strongback a central screw can be used as in the previous cases. The forcing bolt, or eyebolt, is generally used for lifting the piston after it is removed from the rod.

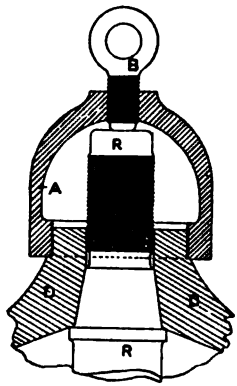


FIG. 139.—Removal of Piston from Rod.

In some cases the rod has been withdrawn from the piston by fitting packing, generally wooden chocks, between the piston and the cylinder end, and then pulling the rod away by turning the engine by hand. This method should not be used, as it brings unknown strains on the cylinder, and may be dangerous.

Piston Packing and Fittings.—The piston, in addition to being made strong and stiff enough to resist deformation under maximum load with minimum weight of material, and being strongly attached to the piston rod, must also be made to work steam-tight in the cylinder with minimum friction. This is arranged by fitting the body of the piston with metallic packing, so that the body itself never comes into actual contact with the metal of the cylinder liner.

The *junk ring* is used to keep the packing in place (*J* in Figs. 138 and 141), and it is secured to the piston by studs and nuts as shown; a *guard ring*, *G*, is fitted to prevent the nuts slacking back. The studs, *Z*, in both the junk and guard rings are fitted with square necks; and split pins are fitted for the guard-ring nuts, to prevent them slacking back. The advantage of fitting a junk ring is that the packing can be examined and fitted without removing the piston from the cylinder. The junk ring should have about 0·005 clearance and should not press hard on the packing, and it should be possible to move the packing ring round comfortably by hand after the junk ring is tightened up in its place. The packing ring should, however, be sufficiently tight to keep the steam from leaking past it, and at the same time it must allow the ring freedom to move up to its working

face on the liner; the sides of the ring are therefore frequently scraped up to a true surface where they butt on the flange of the piston and on the face of the junk ring. The junk ring is made of forged steel (Fig 140); but if, as in Fig. 141, it is allowed to come in contact with the metal of the cylinder liner, it may then be made of cast iron.

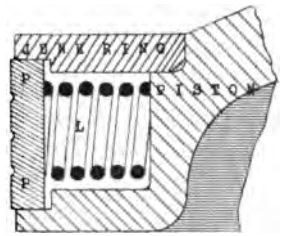


FIG. 140.—L.P. Piston Packing.

The most common form of **piston packing** is shown in Figs. 138 and 140, in which it consists of a single broad ring of cast iron or phosphor bronze, *P*, forced out by a gentle and uniform pressure by springs, *L*, behind it. The springs are now of the spiral type, but coach springs were used in the earlier engines. The packing ring projects radially about $\frac{1}{8}$-inch beyond the body of the piston and of the junk ring. The spiral springs are fitted into recesses, *L*, behind the ring, and the pressure exerted by them on the working face should be about 2 lb. per square inch of ring surface for the L.P. cylinder, and, if used for the M.P. cylinder, about $2\frac{1}{2}$ lb. per square inch is allowed. The springs are about $2\frac{1}{2}$ to 3 inches in diameter, made of wire $\frac{1}{4}$ to $\frac{3}{8}$ inch in diameter; and one spring is fitted at a point midway between each pair of junk-ring bolts, or about 9 inches apart.

The packing ring is cut diagonally (*T* in Fig. 138), and a *tongue piece* is fitted to close the gap, and at the same time allow for expansion of the ring and for wear. The tongue piece is generally fastened to one end of the cut ring by two or more countersunk screws; the other end is left free to move inside the ring and along its edge, as shown.

For the H.P. and M.P. cylinders, as a rule, no springs are used to press out the packing against the liner. The packing rings are made a little larger in diameter than the liner which they are eventually to fit; they are then cut, generally diagonally, and fitted with tongue pieces if required. Sufficient length is cut away from each ring to allow it to enter the cylinder liner, and at the same time leave sufficient space for expansion between the ends of the ring at the cut. The ring itself tends to keep its natural shape, and in consequence springs out against the liner, and is generally steam-tight against it.

Ramsbottom rings fitted to a piston are cut and made in the

way mentioned above, but without tongue pieces. The divisions are spaced around the piston so that no two are opposite, and with very fast-moving engines the leakage is inappreciable. These rings are generally made of hard phosphor bronze.

The more recent types of H.P. and M.P. pistons are fitted with **restrained rings** of hard bronze (Figs. 141 and 142). The

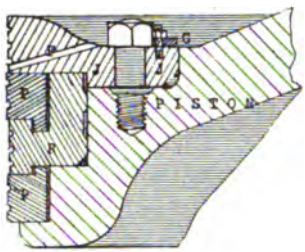


FIG. 141.

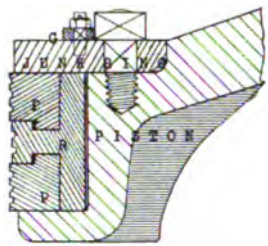


FIG. 142.

Restricted Packing Rings.

tendency of the packing rings, *P*, to spring outwards is restricted to a small amount by a *tension ring*, *R*, made of cast iron. *R* is solid and uncut, with a very slight clearance, $\frac{1}{64}$ to $\frac{1}{32}$ inch, between it and the body of the piston, and about $\frac{1}{32}$ to $\frac{1}{16}$ inch between it and the packing rings, diametrically. The flange of the piston is made so much smaller in diameter that the tension ring prevents it coming in contact with the cylinder liner. The piston in Fig. 141 is fitted with drain holes, *F*, as it is not naturally self-draining.

Fig. 143 shows Lockwood and Carlisle's double-action metallic

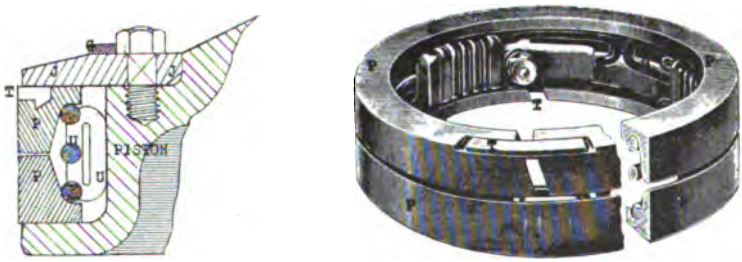


FIG. 143.—Carlisle Piston Packing.

packing rings and springs. The packing rings, *PP*, are strongly pressed away from each other by the continuous spring *U*, to make a steam-tight joint at the faces of the junk ring and the flange of the

piston. Each ring is fitted with a tongue piece, which completes the continuity of contact. The spring is locked, and the rings overlap its edges, which holds them in position and restrains the steam from unduly pressing the rings against the working barrel of the cylinder. When required, the spring can be enlarged in diameter by removing the screw *N*, and lengthening the hole in the plate to allow for the requisite liner, after which the screw is replaced.

This type is also used for cylindrical slide valves, with certain modifications to make it suitable.

Uncut or solid rings are fitted to some small pistons. About $\frac{1}{200}$-inch is allowed between the ring and the cylinder, which is generally made without a liner in small engines; the clearance allowed makes the ring a fairly good working fit when the engine is warmed up to its work. In all cases the rings should be tried separately in the cylinder or working barrel from top to bottom of the stroke, to see that they are a proper working fit, and at the same time can move freely, when worn, up to the working face; this is particularly necessary with uncut rings. The working barrel is generally bevelled off at the top and bottom of the stroke, so that the ring partially overruns the parallel part at each end; this prevents a shoulder being worked up at the ends of the stroke.

Mr. Yarrow has introduced a system of fitting uncut or solid rings on a solid piston. The rings are three in number, and they must be so proportioned that their elasticity is sufficient to get them into position. The rings can accommodate themselves to the shape of the cylinder, which is more or less true of all piston rings, but not of a solid piston, which cannot do so. As no junk ring is fitted, the piston must be withdrawn from the cylinder to fit the rings in place.

Water grooves are now fitted to nearly all packing rings, whether for steam pistons or for pump plungers. These are shown in some of the figures. When a junk ring or tension ring is fitted in such a way that it may come in contact with the working surface of the barrel, water grooves are cut in it also. The grooves should be spaced about $\frac{3}{4}$ to 1 inch apart, and cut from $\frac{1}{8}$ to $\frac{1}{4}$ inch wide and deep, rounded semicircularly at the bottom of the groove. Their object is to form a space where a film of water can be held to assist the lubrication of the packing ring; their usefulness is now well established from some years of practice.

Piston-rod Packing.—Where the piston rod passes through the end of the cylinder, it is necessary that it should be steam-tight, and a packing gland is fitted at this part (Figs. 133 and 183).

Next to the cylinder interior a **neck ring** is fitted, which is a moderately good fit on the rod, and made of gun metal. This forms the bottom of the gland—actually the top in an inverted engine; it is perhaps better to term it the *inner* end. The neck ring is clearly shown in Fig. 144, which is an enlarged view in section of a packing gland and a type of packing made by the *Combination Metallic Packing Coy.*

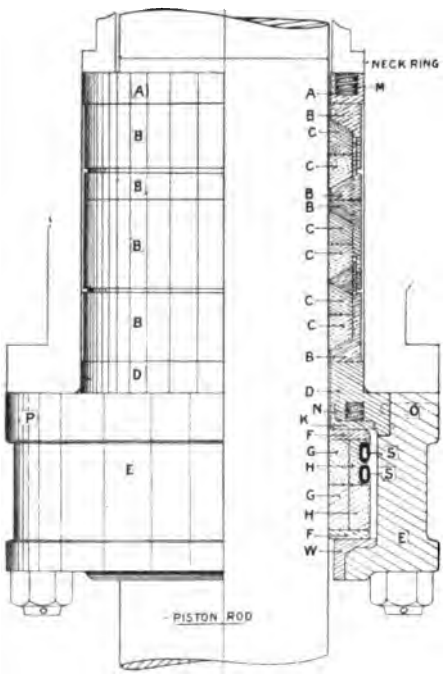


FIG. 144.—C.M. Packing, "H.H." Type.

The packing consists of gun-metal coned and telescopic holding rings, *BB*, which form the seating for the working rings *CC*, made of white metal, and coned in the opposite way. A distance piece, *D*, is inserted to keep the whole of the rings in place, and this distance piece forms the seating for another packing. The metallic packing is pressed on to the rod by spiral springs, *M*, of special shape, suitable for their position, in a spring holder, *A*, next the neck ring. The neck ring in any engine cannot be made a very tight fit, so that a little steam leaks past it and presses on the inner ring, which in its turn keeps the white-metal rings up against the rod, and so relieves the springs of their work after steam once finds its way into the recess between the neck ring and the inner packing ring. The pressure on the rod exerted by the packing is therefore proportional to the pressure in the cylinder, and this may reduce the friction to a minimum. The whole of this packing is contained in hollow metal boxes, and this arrangement permits a better adjustment and fitting of the packing, because it can be inserted complete into the stuffing box.

Sufficient clearance is allowed between each pair of rings for wear and consequent tightening up of the glands. The outer or containing rings, which are made in two halves, are held together by screws in some cases, when it is not necessary for them to touch the stuffing box on the outer rim. This allows a little side play in the rod, and is preferable. If the packing is well fitted and the rod runs truly in line, there is little trouble in keeping a steam-tight joint, and good results are obtained. The white-metal rings are made in three or four parts, and care must be taken that the ends of contiguous rings are not in line, and that they cannot butt when warmed up and the engine is working. No adjustment of the metallic packing can be made when the engines are at work. Water grooves about $\frac{1}{2}$-inch apart are usually cut in the white-metal rings.

The outer packing consists of two rings, *G*, of wearing metal contained within two other rings, *H*, which, by means of springs, *S*, fitted on the outer circumference, press the wearing rings *G* on to the rod. Vertical movement of the rings *G* and *H* is prevented by two face rings, *F*, one above and below, and above the upper face ring is another ring, *K*, pressed downwards on to the packing by spiral springs, *N*. Below the lower ring *F* is a distance angle ring, *W*. The outer packing is held in place by a box, *E*, which is secured to the upper packing box by studs and nuts. Oil can be introduced through a hole, *O*, to the rod and packing, and opposite it a drain, *P*, may be fitted for carrying away any excessive water or pressure—when the engines are standing under steam, for instance.

Mixed Packing for Rods and Spindles.—In many cases, particularly for auxiliary engines, the ordinary metallic packing is not altogether suitable, and soft packings are used. These, although useful in preventing leakage, are not conducive to a good appearance of the spindles, nor are they very efficient from a point of view of frictional resistance. Recently a fairly good packing of the graphite type has been introduced to the naval service, and it requires some discretion in its use. For a rod moving truly in line and in clean condition it is advisable to fit a closely fitting neck ring either of white metal or gun metal, and then the graphite packing is not forced into the cylinder. In some cases the gland may require also to be closely fitted to prevent the packing being forced into the space between it and the rod. If the rod cannot be brought

into line and turned up, it is advisable to fit elastic core packing for the inner and outer turns, with the graphite between them.

For glands of expansion joints, of pipes, etc., graphite may be used in the same manner; and failing graphite, alternate turns

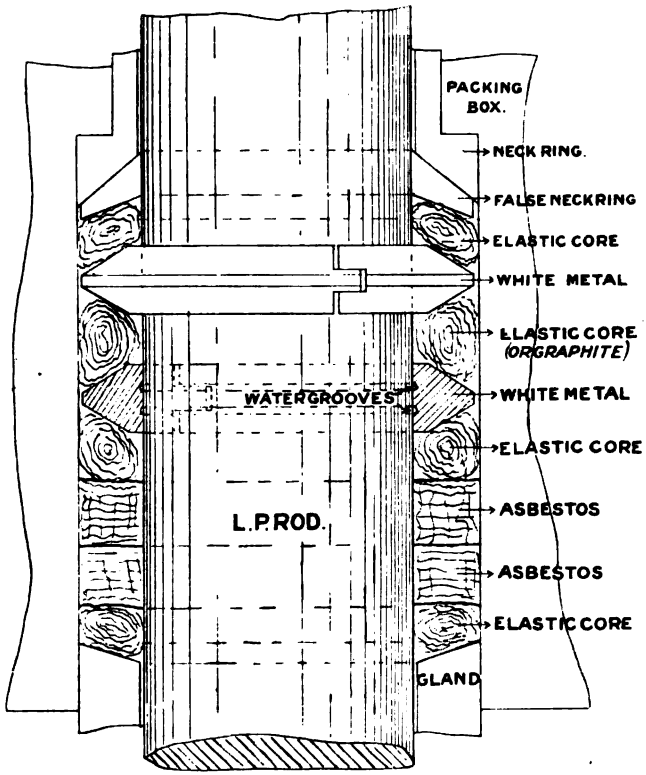


FIG. 145.—Mixed Metallic and Soft Packing.

of elastic core and asbestos, the inner turn being elastic core, makes a good and durable joint.

For large rods, piston rods, and others up to 9 inches in diameter, subject to a vacuum, the author designed and fitted packing of the mixed white metal and soft types, as shown in Fig. 145. This allows for some lateral play of the rod, and by fitting a false neck ring (in halves for very large rods) with a shallow cone, the heavy pressure is relieved from the rod. It appeared to be entirely satisfactory for L.P. piston-rod packing,

and a decided improvement in vacuum and appearance of the rod was obtained with it in comparison with soft packing alone; on the whole, there was some reason to think it superior to a complete set of metallic packing, and it is cheaper. In all cases asbestos packing should be withdrawn from glands if the engine is not to be under steam again within a few days, otherwise the rods are marked and finally become scored.

For the adjustment of soft packing it is arranged that the nuts *A* (Fig. 145A) on the studs can be all tightened up together by means of cogs fitted to each nut, and the whole of which are in gear with a central cogged ring, *B*, fitted to revolve about the gland *C* as an axis. This gearing, *G*, is always made workable from the starting platform. The flange *D* of the gland on which the cogged ring is fitted is cupped on its upper side, into which oil can be run for lubricating the piston rod. Just below is fitted a third gland, *E*, packed with a soft gasket packing, which is used to retain the oil on the rod and to prevent it running away; some, of course, works through this gland, but the greater part is retained.

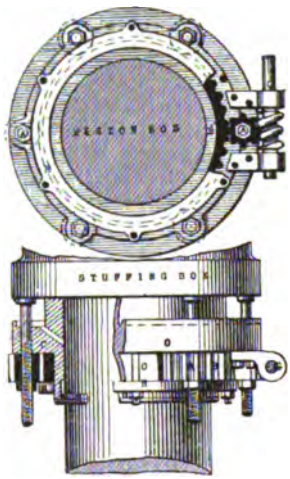


FIG. 145A.—Gland tightening-up Gear.

United States Duplex Packing.—For over 150 lb. gauge-pressure a double packing is more efficient, and the type shown in Fig. 145B is frequently fitted. The packing nearest the piston is of the white-metal coned type shown in Fig. 144, and tends to wire-draw the steam, while the block packing below it ensures steam-tightness.

A cupped plate, *A*, supports springs *B*, which press on the flange of a plate *C*. (In some cases *C* is prolonged upwards, and then forms the neck ring.) *C* presses on a coned gun-metal ring *D*, which compresses the coned white-metal rings *EE* (each cut in three parts) on the rod. The (rectangular section) white-metal ring *F* tends to block the passage of steam at the nearly butting ends of the parts of the rings *EE*. The rings *D*, *E*, *F* are contained in a gun-metal casing *G*, pressed down on *K*, which makes a ball joint with the socket piece. A clearance of about

$\frac{1}{32}$-inch between *A* and *C* allows a slight leakage into the space about *G*. A drain is fitted to this space for occasional use.

The block packing, which is alone necessary for moderate pressures, is compressed between the two plates *MM* by vertical springs fitted in the socket plate, and pressing on the plate *L*, which has a projecting lip. This lip prevents the possible fracture of the springs by limiting the side play. The packing consists of four white-metal blocks *Q*, each contained in a gun-metal case *R*, which are pressed directly on to the rod by spiral springs *B*. The blocks are fitted in pairs on opposite sides of the rod, one pair above and at right angles to the other. Each pair works between gun-metal guide blocks *N*, which are pressed in by spiral springs. The packing and guide blocks, with their springs, are contained in two nearly similar casings *P*, and dowel pins are fitted to prevent their rotation relatively to each other. A ball-and-socket joint is made between *K* and the casing. Some steam leaks through the upper packing, and can fill the space about the lower blocks, as access is allowed through a clearance space (about $\frac{1}{32}$-inch) at the lip of *L*. This pressure assists the springs to press the packing blocks against the rod.

The action of the coned packing has already been described. It is seldom entirely steam-tight, and a second packing is generally fitted to save the water which dribbles past it. With soft packings and many metallic packings, tightness is only obtained by hand screwing up, which produces an unknown pressure on the rubbing surface of the rod. This pressure is constant at all parts of the stroke, and produces an unnecessary frictional resistance.

On the contrary, the springs behind the packing blocks *Q* exert a pressure of about $1\frac{1}{2}$ lb. on each square inch of rubbing surface. Each block embraces about one-third the circumference of the rod, making in all $1\frac{1}{3}$ times the circumference. The depth of the packing is from $1\frac{1}{4}$ to $1\frac{1}{2}$ inches. The pressure exerted by the block packing on the rod varies with the pressure obtaining access from the cylinder to the space between the casing *P* and the blocks, and is in consequence never greater than this amount *plus* the definite pressure exerted by the springs; in other words, the packing is *steam-setting*, and the frictional resistance is a minimum.

The ball-and-socket joints allow a moderate lateral play of the rod, which is a defect common to all rods either from original mal-alignment or from wear. Space is allowed throughout for

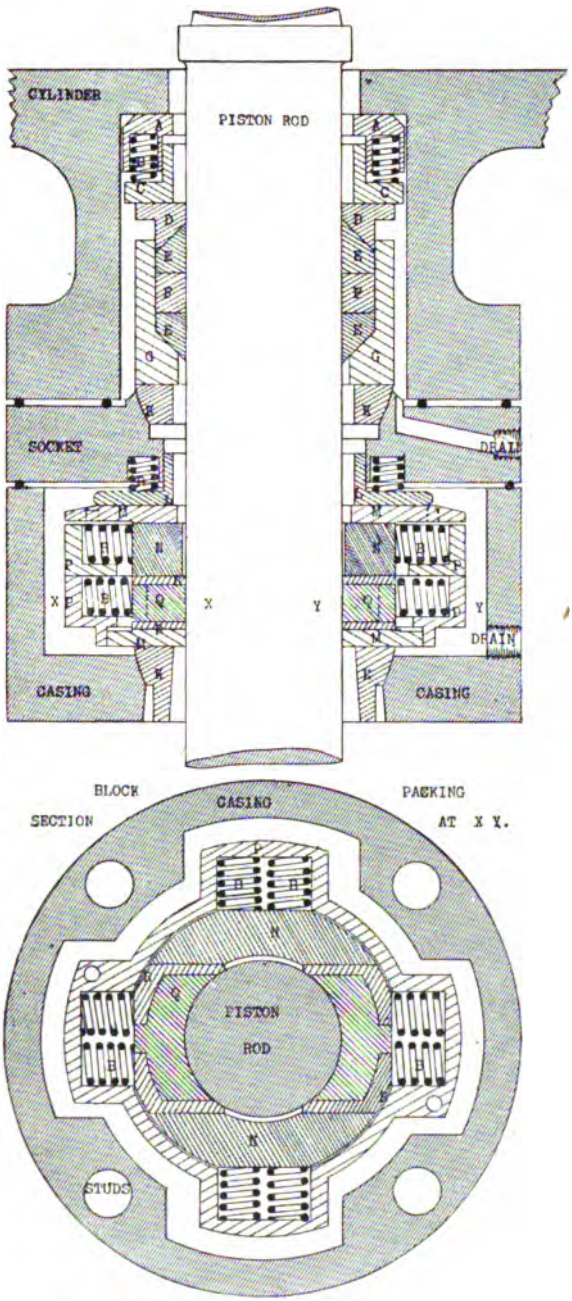


FIG. 145B.—U.S. Piston-rod Packing.

this lateral play, and when the rod is made with a collar under the piston the space is conveniently used for passing the collar through it. Stronger vertical springs are fitted for use with a vacuum; 16 lb. pressure per square inch is allowed for the L.P. cylinder, and prevents the air passing through the socket joints into the cylinder. The joints of the socket and cylinder stuffing-box are sometimes made with wire-ring joints, as shown in the sketch.

Packing of the block type has been working for many years on land and locomotive engines without any outlay on repairs. It has all the elements of a successful packing, and efficiency can be obtained with it when properly fitted and adjusted.

Slide-valve rod packing is generally of the same description as piston-rod packing.

CHAPTER XV

THE CONNECTING RODS AND SHAFTING

Piston Rod.—The connection of the piston with the piston rod has already been described. The rod itself is now generally ground, after turning, to produce perfect parallelism. The connection of the lower or outer end of the piston rod with the connecting rod is made in two ways, as shown in Fig. 146.

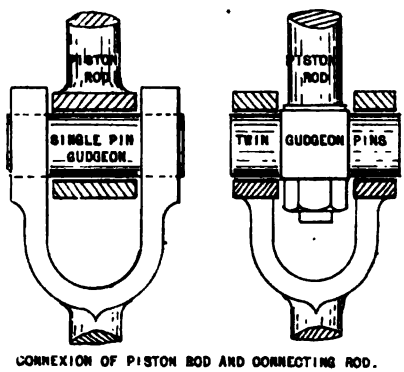


FIG. 146.—Gudgeon Connections.

1. The top end of the connecting rod is forked, and a single gudgeon pin is fixed through the two ends of the fork. A bearing is fitted in the end of the piston rod, as shown in Fig. 147, and embraces the gudgeon pin.

2. The lower end of the rod is made with a coned end, similar to the one fitted for the reception of the piston, and this cone fits into a cross piece, or *crosshead*, the ends of which form twin gudgeon pins. Bearings are fitted to the top ends of the fork, and these embrace the gudgeon pins, as shown in Fig. 150.

The pressure exerted on the piston is communicated to the rod, which is alternately in compression and tension. By considering it as a column under compression, the size required generally covers that necessary for the tensile stress. From the piston rod the pressure is transferred to the connecting rod, and so to the crank. The piston and connecting rods are not always in line, and a proportion of the pressure is transferred to a guide, as explained in Chapter XIII. The end of the rod is fitted to transfer the

pressure to the guide, and is thus relieved of the bending action to which it is otherwise subjected.

Guide.—The back column is made suitably to receive the guide, or shoe, in which the slipper attached to the end of the piston rod moves. The mutual action of the guide and slipper prevents any bending of the piston rod, which is due to the obliquity of the connecting rod. The guide is generally made separately from the column to which it is bolted, as shown in Figs. 127 and 149. A space left between the column and the guide is made water-tight, and used to keep the ahead rubbing surface cool by a continuous circulation of water through it, preferably in an upward direction. The outlet for the circulating water is generally led into the crank pit, so as to be visible from the starting platform. In destroyers the circulating water is led overboard, above the water-line, so as to be visible. Sufficient head of pressure is obtained for this purpose by taking the supply from the main circulating-pump discharge.

A fairly accurate measure of the heat and work wasted in friction at the guide can be obtained by observing the rise of temperature of the outlet water and the weight used in a certain time. Hand-holes are provided behind the guides, so that the temperature of the back of the rubbing surface can be felt while the engines are working.

The astern guide surface is also made separately from the column, and generally from the ahead surface, one strip on each side of the slipper, and these are bolted on to the front of the ahead guide, as shown in Figs. 127 and 149. The area of the astern surface is about three-fourths that of the ahead surface, and is sufficient for ordinary practice. The guides for both ahead and astern working are made of cast iron or steel, and suitable grooves are made in them for distributing the lubrication over the rubbing surfaces.

In some cases, generally in mercantile practice, the front and back columns and guides are identical. A special type of cross-head is then required, and one used for this purpose is generally fitted for twin gudgeon pins. The double guides, as described, are heavier, and when it is considered that a sea-going ship is seldom required to go astern for long periods, the open-fronted engine appears to be suitable and preferable.

Slipper.—For large engines the slipper is usually a separate fitting, because it is difficult to make in one piece with the cross-

head, and because it also permits adjustment for wear. Various types, all very similar, are shown in Figs. 147, 148, and 149.

In Fig. 147 a horizontal dovetailed groove is cut in the slipper

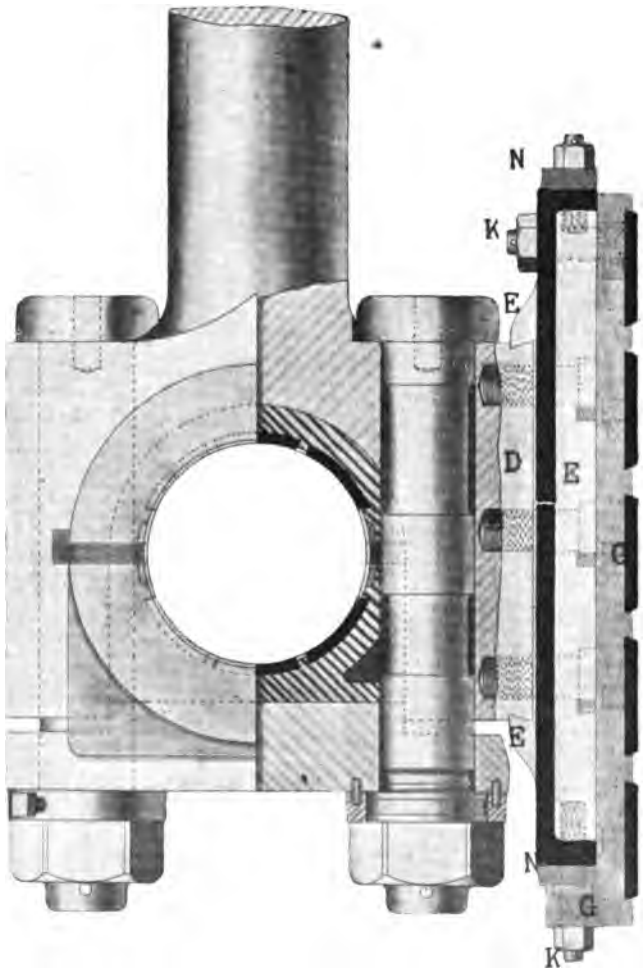


FIG. 147.—Slipper and Single Gudgeon-pin Bearing.

EE, which fits tightly over a fillet, *D*, on the crosshead; and the two parts are held together by sunk-headed screws, which are secured from slacking back by the set screws shown partly embedded in the heads of the screws. The ahead rubbing surface of white metal is

contained in a gun-metal liner *G*, which is removable without disturbing any important parts of the machinery. The liner *G* can be slid upwards into its place, and when the surface is worn, a thin sheet liner is inserted between it and the slipper *E*, to restore the

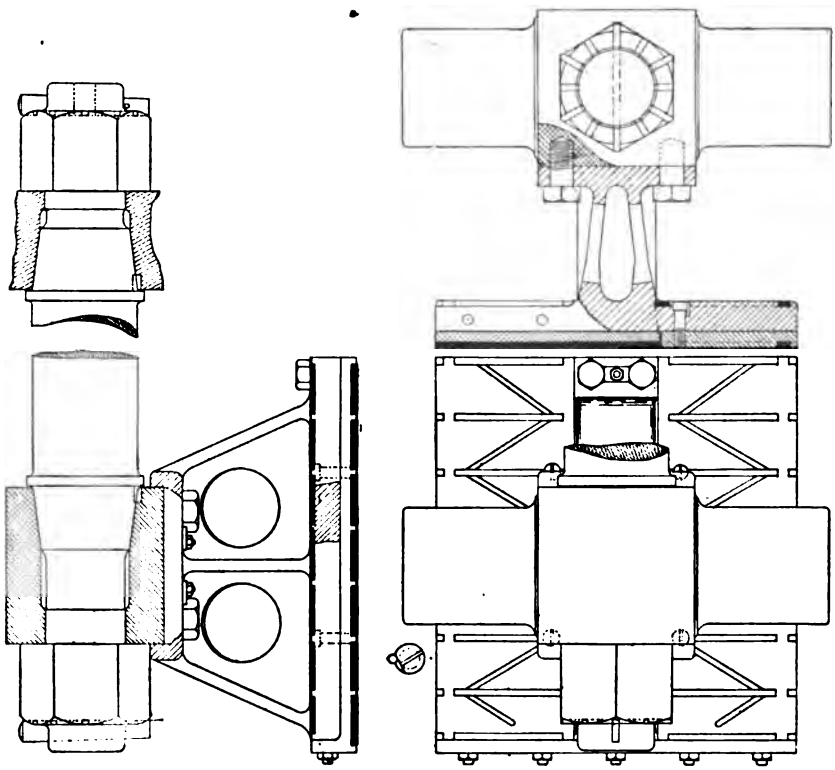


FIG. 148.—Crosshead and Slipper.

rod to its alignment with the axis of the piston and cylinder. When in place the liner is held by studs, *K* (fitted with square necks, collars, and the usual split pins, to prevent slacking), and the holes in the liner are elongated, to allow for adjustment. These studs are not quite safe under the tensile stress to which they are subject, and additional studs, *K*, are fitted through the slipper and liner at the top end, with the usual nuts and stops to prevent their slacking off. The white metal is dovetailed into the containing liner, and cross spaces are left in it to retain the

oil on the rubbing surfaces, and to distribute it over the surface evenly.

The white metal which forms the astern rubbing surface is generally dovetailed into the slipper without any containing gun-metal liner, as shown in Fig. 149. But in a few cases the astern

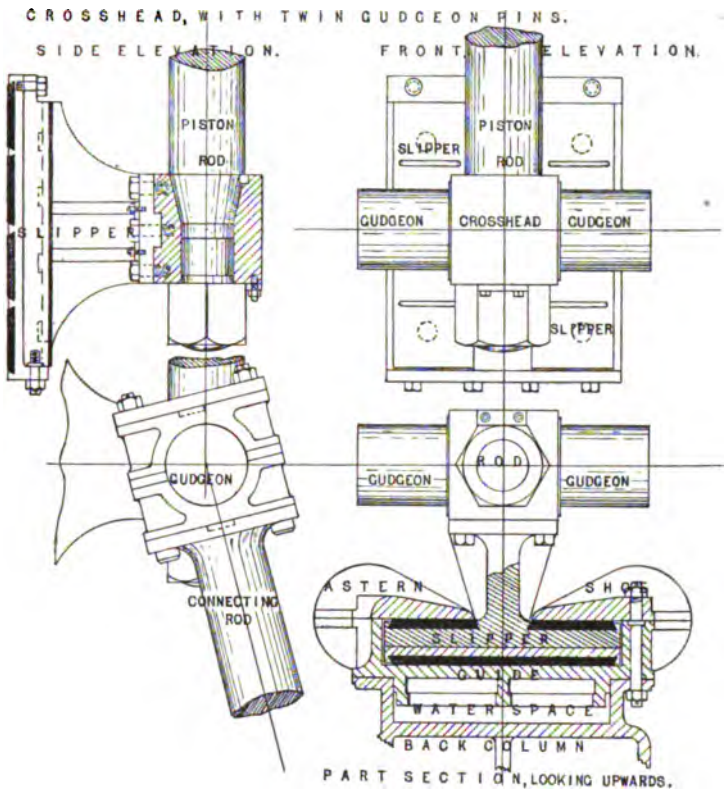


FIG. 149.—Twin Gudgeon-Pins and Slipper.

rubbing surfaces of the slipper are detachable (as shown in Fig. 147 for H.M.S. *Drake*), made of white metal, without containing pieces, and inserted from the top and bottom ends of the slipper. Flanges above and below the four strips shown turn inwards, and next these flanges gun-metal strips, *N*, are fitted to receive the nuts of studs fitted through the strips and flanges into the ends of the slipper.

In good practice the recesses for the white metal are machine

cut, and the white metal, after it is run into them, is machined to shape, and the recesses for lubrication are machined out. White metal is usually fitted for both ahead and astern surfaces of the slipper.

Under ordinary conditions the wear on the astern surface is inappreciable, and therefore the rod tends to retain its alignment when going astern; but if the extra liner, instead of widening the difference between the two rubbing surfaces, merely sets the rod end *away from the ahead surface*, it at the same time pushes the astern surface of the slipper away from its guiding surface, and produces mal-alignment. For this reason the adjustable liner should always be fitted between the ahead and the astern rubbing surfaces of the slipper, and *not* between the crosshead and slipper.

Connecting Rod.—The connecting rod is made of forged steel, and there are two principal types.

With the inside or *single gudgeon-pin* type (Fig. 147), if a collar is fitted on the piston rod under the piston, it may necessitate the neck ring, stuffing glands, and other fittings embracing the piston rod (which are of smaller internal diameter than the external diameter of the collar) being cut in segments. This conduces to leakage at the glands, unless special precautions are taken, and the result is not so satisfactory a mechanical fitting as the *uncut* glands, etc. The forging and machining of the piston-rod end is also more cumbersome than the outside or *twin gudgeon-pin* arrangement (Fig. 150), now frequently adopted. It is so proportioned that the connecting rod oscillates just clear of the piston-rod nut at the fork end. Two adjustable bearings are fitted, one to each of the twin gudgeon pins, and properly secured to each fork end. In this arrangement it will be noticed that the *bearings* are carried by the connecting rod, which is the reverse of the previous type, in which the *pin* is carried by the connecting rod.

The length of the connecting rod from centre to centre of its end bearings is almost invariably made equal to four cranks—that is, to twice the length of stroke of the piston. Its minimum diameter is calculated by considering it a pillar in compression, which generally covers its necessary strength considered as a rod in tension. A bending moment due to the centrifugal action of the rod has also to be considered, and to meet this, the rod is tapered, with the larger diameter at the crank end. In very slow-moving engines the rod was made with its largest diameter at about its mid-length, but with the quicker-moving engines of the present day the

largest diameter has gradually fallen to the part immediately above the crank.

In some of the older ships, where gun metal was used for the top-end rubbing surfaces, the gudgeon pin was made of wrought

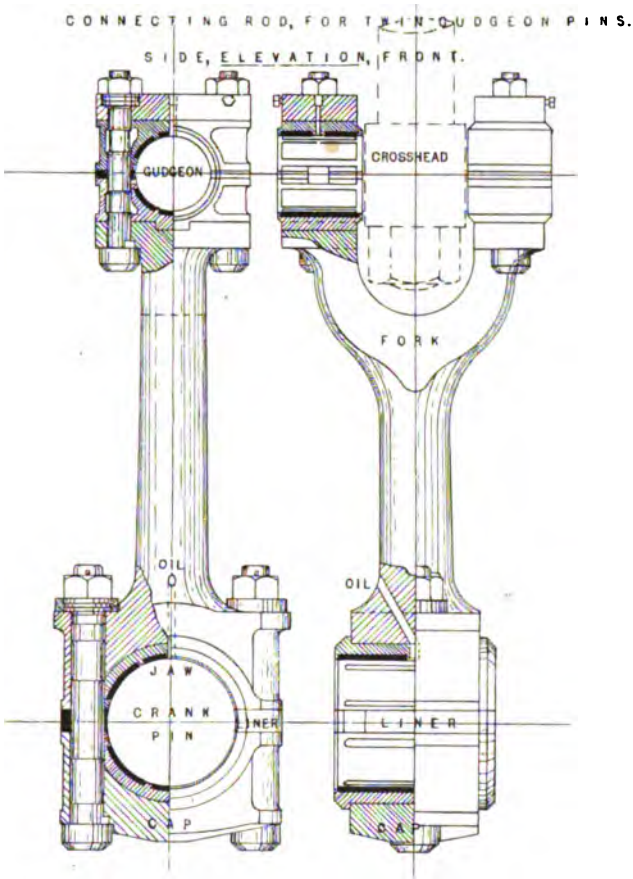


FIG. 150.—Connecting Rod for Twin Gudgeon-Pins.

iron, case-hardened on its surface; but with the white-metal linings, now almost universally adopted, mild steel pins are used. The twin pins are forged in one piece with the crosshead. The single pin is shrunk into the holes in the fork, and set screws are screwed in at the junction to prevent the pin turning; when worn oval it

can be trued up by filing or lapping, and is sometimes turned round a quarter-turn so as to bring the oval the opposite way.

Connecting-rod Bearings.—Although there is some considerable variation in outward form and shape, the same general principles are apparent in all connecting-rod brasses. They are made in two parts, upper and lower; and for convenience the bolts at the top end point downwards when a single gudgeon-pin is used, but for twin pins and all crank-head brasses the nuts are conveniently fitted on top. The outer or cap brass is stiffened by a steel cap (Fig. 150), and liners are fitted between the two parts, so that by reducing their thickness the brasses can be brought together, and thus embrace the pin more closely. The diametrical working clearance between the pin and the brasses should be about 0·1 per cent of the diameter of the pin, but much depends on the perfection of workmanship and the alignment. (See also Chapter XXX.)

All the bearings are now lined with white metal in strips, which are fitted longitudinally to the pin; but the upper and lower edges are usually made of the shape shown in Fig. 339. This arrangement conduces to a better distribution and retention of the lubricant. The white-metal lining extends over about two-thirds of the semicircumference at the top and bottom of a pair of brasses, with the clearance at the distance pieces (liners).

In the more recent ships an *oil-containing strip* of white metal extends through the brasses and liners, so as to completely embrace the pin as near each end as possible; this is to prevent the oil working out of the bearings too quickly. A strip is shown in the part section of the crank-head brasses in Fig. 150, and strips are also used for main-bearing and gudgeon brasses.

The *connecting-rod bolts* are made of steel, and now generally fitted with wrought-iron nuts; the bolts are made with stretching lengths just inside the nuts, and stops are fitted under the heads, to prevent them turning in their holes. A little consideration shows that the bolts in the connecting rod and main bearings are only subject to a tensile stress, and are never in compression.

For the proper adjustment of the connecting-rod brasses, the opposite end of the connecting rod should be disconnected and the rod swung through the working angle, and adjusted until it swings fairly in line. This swinging adjustment is particularly necessary with twin gudgeon-pins. For the lower or crank-head brasses the

rod should be swung at various positions of the crank to completely test the adjustment.

Main Bearing.—The crank shaft rotates in the main bearings, and one is fitted on each side of each crank as closely as possible, to reduce the bending moment on the shaft, as shown in Fig. 154.

The main-bearing brasses are white-metal lined, and the oil-containing strips are fitted near the ends of the brasses, completely embracing the shaft, as in the crank-head brasses. The lower brass is made semicircular (Fig. 151), so that when the steel cap, top

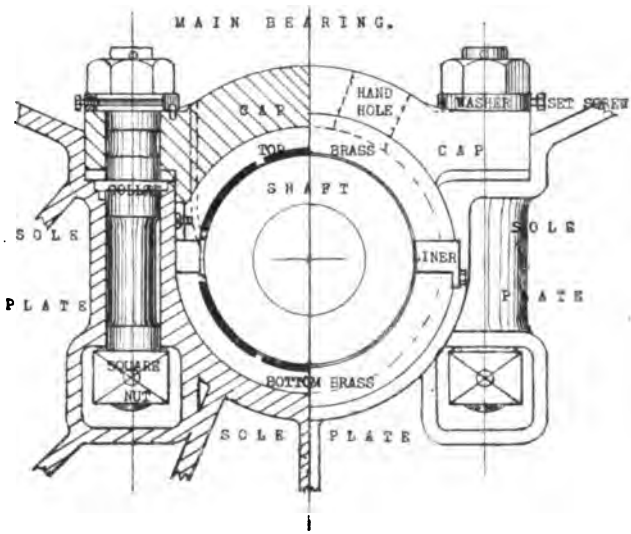


FIG. 151.—Main Bearing.

brass, and liners are removed, it can be turned round on the shaft until it is bottom upwards, and can then be lifted away for examination and repair; thus without lifting the crank shaft the bottom main-bearing brasses can be removed and refitted. A hand-hole is made in the cap, so that the top brass can be felt when the engines are working. Steel bolts fitted with a collar have been substituted for studs securing the main bearings in comparatively recent years. Studs have an unpleasant habit of slacking back which cannot be easily detected. The lower end of the bolt is fitted with a wrought-iron nut, generally square, secured by a pin through it; the bolt is prevented from turning by a stop under the collar; a stretching length is fitted inside the cap, as shown, and the

usual wrought-iron spigoted nut, washer, dowel pins, set screw, guard plate, and split pin are fitted.

In addition to the lubricator fitted for the top of the top brass, another is now fitted, and conducts the oil to the lubricating hole, shown in dotted lines in the figure, leading to the middle space about the liner for the independent lubrication of the bottom rubbing surface.

Centrifugal Lubricator for Crank-pin Bearing.—The crank pin is usually made hollow, and is then used for lubrication, as shown in Fig. 152. The oil runs down a small pipe, *A*, into a

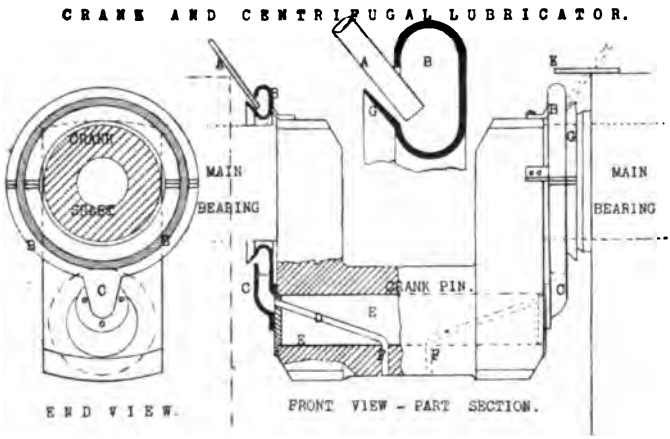


FIG. 152.—Centrifugal Lubricator and Crank.

double groove *B*, which is fitted on to the side or web of the crank, and revolves with it. The centrifugal force given to the oil by the rotation of the shaft drives it into the pipe *C*, leading into a pipe, *D* (inside the hole in the pin *E*), and thence through a hole, *F*, drilled through the pin and leading to the bearing surface. Two such double grooves are fitted to each large crank, one on each side. It has been found that when water is used on the main bearings it finds its way into the crank bearing, and to prevent this happening, a lip *G* is now fitted, as shown in the figure, and also an enlargement, which throws the water off. This lip allows dirt to fall into the double groove, and over it a plate, *K*, is fitted on each main bearing to protect it. In all cases a wire-gauze screen is fitted at some part of the outlet pipe into the crank, generally at the joint between the outlet pipe *C* from the double groove and the

end of the crank. The centrifugal lubricator must be made in two semicircular parts, so as to get it in place. Lugs are fitted to it, and make the connection with the crank. Oil-ways are cut as necessary in the crank-pin surface.

Forced Lubrication.—In several modern warships fitted with reciprocating engines a system of lubrication under pressure has been adopted with fairly satisfactory results. The crank pits are arranged (see Fig. 153) so as to form a well for the oil dripping

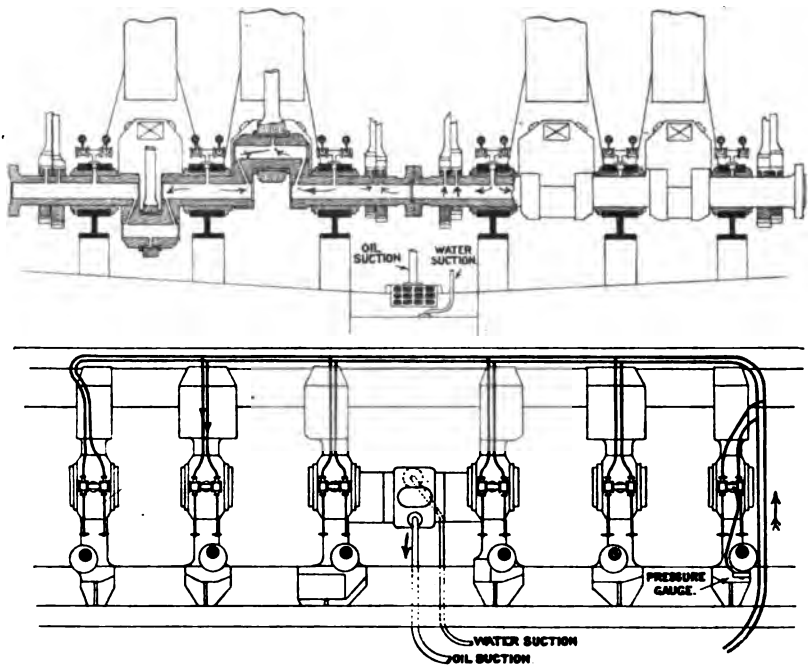


FIG. 153.—Forced Lubrication fitted in a Battleship.

from the bearings, and which drains naturally into the oil-pump suction, which is fitted as a filter. The oil is cooled on its way to the pump by means of a cooler—the water suction for which is shown in the sketch. The cooler is of the surface type, and the oil and water do not mix. One engine generally performs the separate duties of pumping water through the cooler and of pumping oil under pressure into the bearings. Oil under pressure is forced by the pump through pipes, shown by thick lines in the plan, into the main bearings, from which it flows through holes in the shaft

into the inner space of the shaft. From this inner space the oil is forced through small holes into the interior of the crank-pins and so to the bearing surfaces of the crank bearing; a small hole leads to each eccentric bearing surface, and in some instances a hole in the interior of the connecting rod conveys oil to the crosshead bearing. For forced lubrication it is generally necessary to use mineral oil only, because olive oil, although a superior lubricant, tends to saponify and choke the filter, and this tendency is increased by any mixture with either fresh or sea water. A pressure-gauge shows the pressure in the pump delivery; the pressure should generally be regulated to suit the power of the engine; in round numbers 10 lb. per square inch should be allowed for each one-fifth of the full power, but in this as in other matters of lubrication practical experience is the best guide, and what may be successful and satisfactory in one case may not necessarily be so in another engine room. Oil-containing strips are necessary to a successful system of forced lubrication, and careful fitting and adjustment of the bearings helps considerably. The figure shows a duplicate system of supply pipes.

Propeller Shafting.—In Fig. 154 a plan and elevation of the shafting of a twin-screw ship is shown in general outline, with the positions of the various fittings and bearings. The arrangement of each engine and shafting is similar, but fitted on opposite sides of the keel. The two engine rooms are divided by a middle-line bulkhead, and each engine is made so as to be entirely independent of the other.

The shafting shown is for the four-crank, four-cylinder, three-stage expansion type, with two L.P. cylinders. The crank shaft is made in two or four lengths, and to some extent these are interchangeable, and for this reason a coupling, *C*, is frequently fitted at the forward end of the forward length. The adoption of varying angles between the cranks, as used in the Yarrow-Schlick-Tweedy and other balancing systems, sometimes prohibits entire changeability, which is, however, always desirable. At the positions for the eccentrics *D* the shafting is enlarged, so that the cutting of the keyways for securing the eccentric sheaves on the shaft shall not weaken it locally. For each coupling there is a fillet at one end fitting accurately into an opposite socket; this is to maintain a perfect alignment of the shaft. The coupling bolts are a driving fit into their holes, and frequently, to shorten the

length of the shaft, and so bring the bearings closer together, the heads of the bolts are coned, and wholly or partially sunk into the coupling flange. Cone-headed bolts should not be used for couplings subject to propeller or end thrust. The sketch shows the outline of the sole plate into which the main bearings are fitted.

Just abaft the after coupling the thrust shaft *T* is fitted, and next abaft is a plummer block, *P*, frequently incorporated with the thrust seating. Then comes another coupling at the after end of the thrust shaft for coupling with the intermediate shaft, and to this coupling the turning wheel is attached, if there is sufficient room for it; but if not, then it is fitted at *W*, or as convenient. The intermediate shaft projects into the engine room from the shaft alley, and a hole sufficiently large for its withdrawal is cut in the bulkhead at the fore end of the alley. This hole is covered with two semi-circular plates, bolted on to the bulkhead and fitted with a packing gland, *F*, so that the bulkhead can be made water-tight. Access from

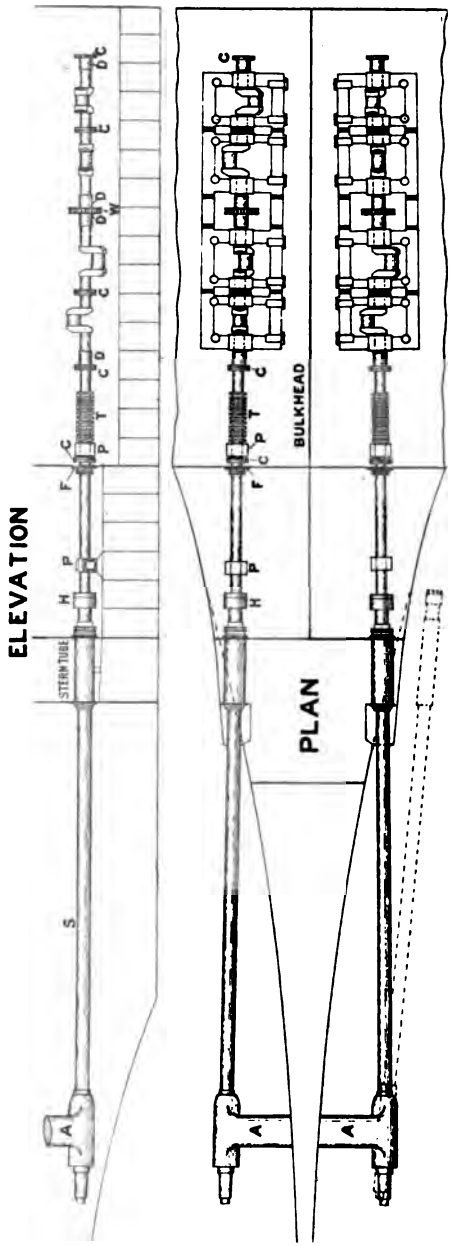


FIG. 154.—Shafting and Bearings.

the engine room to the shaft alley is provided by means of a water-tight door, which can be closed or opened both from the engine room and from some point above the water-line.

The intermediate shaft is supported by plummer blocks, *P*, and at its after end is provided with a coupling suitable to the loose coupling *H* at the fore end of the stern shaft *S*. In some smaller ships the shaft alleys form part of the engine rooms; but as the length of ships is increasing, they now generally form separate compartments. In many of the older naval vessels there is an outside coupling abaft the loose coupling, which is sometimes protected by a partially removable tubular casing, the after end of which is secured to the stern, or "A," bracket. A better flow of water to the screw is obtained in later vessels by omitting the external casing and outside coupling; a coned piece is then fitted in front of the "A" bracket, rising from the shaft, to give an uninterrupted flow of water to the screw.

Plummer Block.—The main bearings support only the crank shaft, and the weight of the propelling shaft must be supported at suitable distances to maintain proper alignment. Unlike the main bearings, which have to take a stress upwards as well as downwards, the plummer block has only the dead weight of the shaft to support; only the lower part of the block is therefore lined as a bearing with white metal, and the upper part is merely a cap, and does not come in contact with the shaft. The cap is used to keep dust and dirt out of the bearing, and is made with the necessary fittings for holding the lubricator and water-service pipes. In Fig. 156 a plummer block is shown incorporated with the thrust block at its after end. If the thrust block is not sufficiently close to the after-crank bearing, another plummer block is fitted at the fore end of the thrust block. It is important that the latter should not be burdened with the additional duty of supporting the shaft. The plummer blocks, as independent fittings, are supported on two or more frames of the ship, so as to distribute the weight of the shaft.

Thrust Block.—When the propeller is rotated it exerts either a pull or a push on the propeller shafting, by which the ship is moved. Some means must be provided for transferring this force to the ship itself, or thrust would be produced on the engine and framing. With screw propellers this force is transferred through the medium of the thrust block, of which a simple type is shown in Fig. 155. On the shaft several collars are formed, and these fit

into recesses accurately made for them in the block. This block is white-metal lined (shown in black section in figure), and there is radial clearance (about $\frac{1}{8}$-inch in large engines) between the recesses and the collars, so that the weight of the shaft shall not rest on the block, which is provided to take the thrust only. The top and bottom parts of the block are generally made hollow, and water can be circulated through the spaces so formed to keep the bearings cool. A large proportion of the power developed must be dissipated in friction at the collars, and for this reason water service is usually necessary to carry away the heat thus generated. The block is securely bolted down to a seating plate, which rests on several frames of the ship, to which it is secured, by means of longitudinal

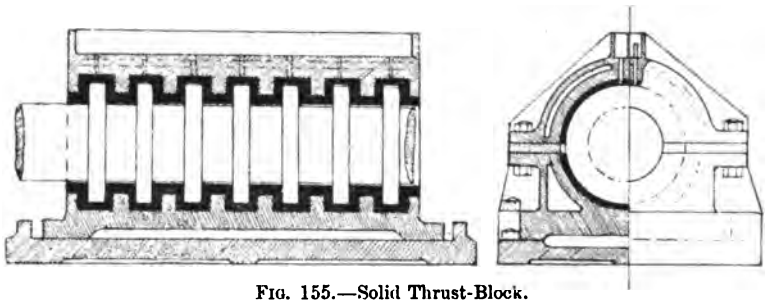


FIG. 155.—Solid Thrust-Block.

tie beams and girders, so as to distribute the thrust pressure over a sufficiency of the plating and framing of the ship. This block can be adjusted to a moderate extent by means of set screws and taper keys fitted between it and the raised ends of the seating. The bolt holes at the bottom of the block are elongated for this purpose, and generally filling pieces are fitted alongside the bolts after the block is properly adjusted, to prevent it shifting on its seating.

The so-called *solid block*, as described above, has the advantage of simplicity, and all the collar surface is effective; but generally the shaft must be lifted to refit the lower part. To overcome this difficulty, Mr. Seaton invented a block which parts vertically instead of horizontally, and it has some other advantages. Previous to the general acceptance of white metal as a good bearing surface, half-rings of gun metal were sometimes used, which fitted into recesses on each side of the collars in the block. These could be driven out with a bent punch without lifting the shaft.

Horseshoe Thrust-Block.—For large ships and great powers

a more elaborate and also useful type of block is adopted, and is shown in Fig. 156. In addition to the ordinary advantages, this block has now an arrangement by which the water used for cooling the bearing surfaces is not allowed to come into actual contact with them.

Five sketches are shown. The left-hand top one shows an end view just in front of one of the thrust-shaft collars; the right-hand top shows an elevation in part section; the lower right shows a plan with the shoes removed and with the thrust bars and nuts for adjustment; the mid left shows a section across one of the horseshoe rings with the space for the internal water service and its connections; while the bottom left shows a plan of one of the rings.

REFERENCES TO FIG. 156.

A, Steel thrust shaft, generally hollow.

B, Collars on shaft *A*.

C, Cast-steel frame, securely bolted down to ship's framing, which takes the propeller thrust.

D, Part of *C*, which forms the lower part of the plummer block at after end of the thrust shaft. If the thrust block is not close enough to the after-crank bearing, a plummer block is also fitted at the forward end.

E, Steel-screwed thrust bars, secured to framing *C*.

F, Steel removable caps, fitted over ends of thrust bars to secure them down to the framing *C*.

G, Wrought-iron nuts at end of bars, used to take the thrust, and also for adjustment.

H, Horseshoe thrust rings, lined with white metal on both sides next thrust collars on shaft.

J, Hollow space in each shoe, through which water is circulated to keep the bearings cool.

K, Oil ways, and oil groove in white-metal lining for **ahead** thrust.

L, Oil way for **astern** thrust.

M, Air outlet, fitted with a cock, from circulating water.

N, Bolt holes for holding down each shoe to frame *C*. Holes in shoes are elongated, to allow for adjustment fore and aft.

P, Phosphor-bronze check nuts, each side of each shoe, on screwed thrust bars *E*, for adjustment of each shoe independently.

R, Guards to prevent nuts slacking back.

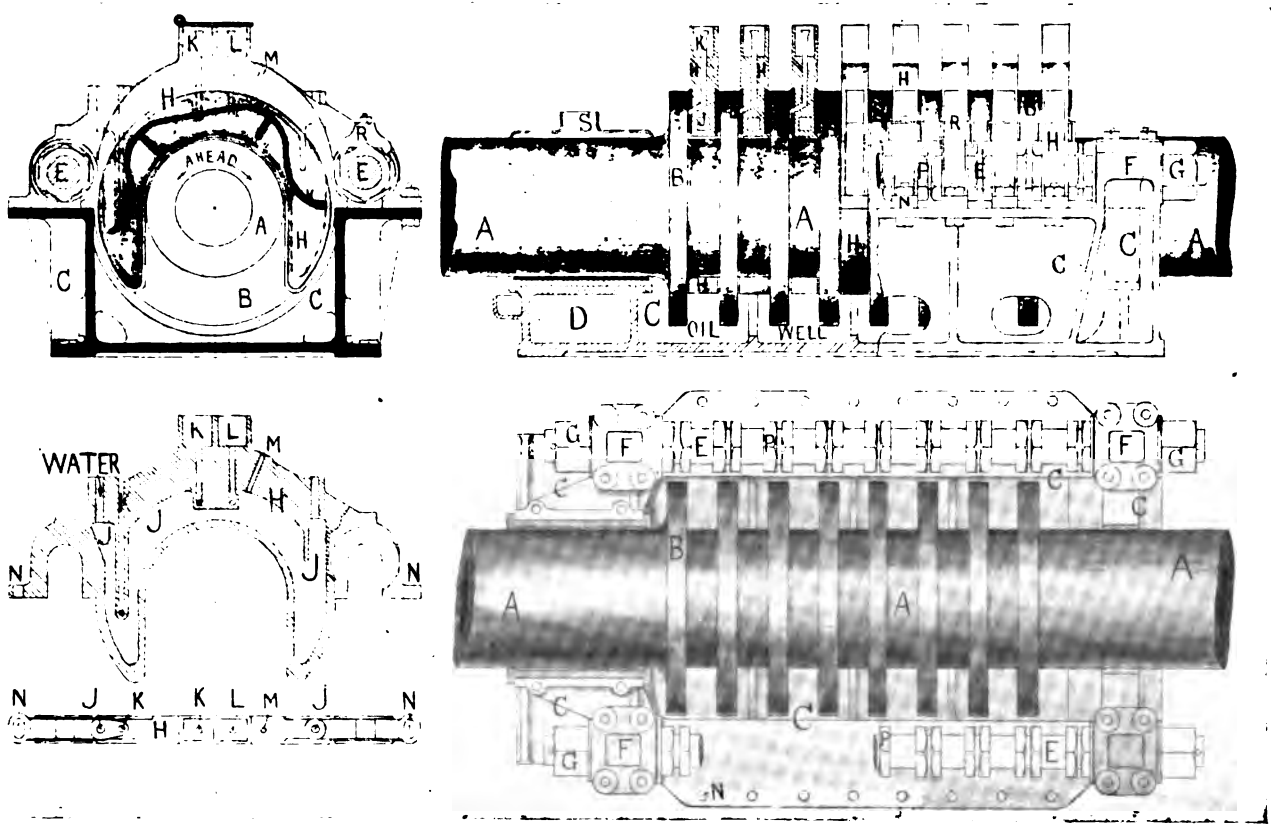


FIG. 156.—Horseshoe Thrust-Block, with Internal Water Circulation.

The advantage of the separate rings or horseshoes is that each ring can be adjusted or renewed independently, and this can be done very quickly; less spare gear is required, and a fair speed can be maintained with only a few rings in effective operation. With all the rings in one containing casting, it is probable that they would all give way together, and the ship might become disabled for a long time. With separate rings, it is possible to renew them one at a time while the engines are at work.

The centre of pressure on the rings is above the centre of the shaft, and it is arranged that the centre lines of the thrust bars *E* are level with the centre of pressure. There is, in all thrust blocks, a difficulty in getting the lubricating oil to remain on the rubbing surfaces, and a special cutting of the oil grooves on the rings is the result, as shown in Fig. 156. It will be noticed that there is an additional hole conducting the oil to the lower part of the shoe, and that the rotation of the shaft in the ahead direction tends to take the lubricant with it, both on its surface and in the groove which assists in supplying it. The oil that escapes from the surfaces fall into the bath, or well, at the bottom, and is again drawn up by the collars, which are partially immersed in it. A single lubricating hole at the top of each collar serves for the astern surface.

The exclusion of the sea-water service from actual contact with the rubbing surfaces conduces to maintaining great and continuous efficiency. Water produces corrosion, then a roughness, which produces a great frictional loss. Good and efficient lubrication prevents rubbing surfaces coming into metallic contact by interposing a film of oil between them; this is the object of all lubricating arrangements.

In several of the most recent warships an improved method of lubricating the thrust collar surfaces has been adopted. Instead of, or in addition to, cutting grooves in the white metal of the horseshoe collars, other grooves, about $\frac{1}{4}$-inch wide and $\frac{1}{8}$-inch deep, *well rounded at the cutting edges*, and extending to within 1 inch of the edges of the collars, are cut, nearly tangentially to the shaft, in the faces of the shaft collars, as shown in Fig. 157. The collars are always partially immersed in oil contained in the well formed by the lower part of the block. Some oil is caught up on the bearing surfaces of the collars, and instead of being thrown off by centrifugal action is retained by the grooves and carried from them

to the surfaces. A very efficient lubrication can be obtained in this way with greater simplicity and less chance of the grooves choking than when the latter are cut in the white metal. The grooves must necessarily be cut in the opposite directions for the *ahead* and *astern* surfaces respectively.

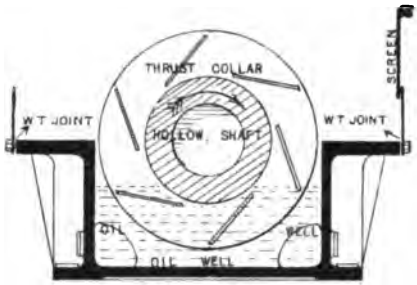


FIG. 157.—Oil Grooves in Shaft Collars.

In H.M.S. *Drake*, considerable trouble having been experienced in the thrust blocks from excessive heating and consequent wear from the use of the sea-water cooling service, grooves were cut in 1906, under the author's direction, as above described, and the lubrication at moderate speeds was improved, but heating was still prevalent at high powers. The oil well was then built up, as shown in Figs. 157 and 158, to save the lubricant thrown off the shaft collars by centrifugal force; and to maintain a constant level in the well, fresh water and oil is now added as required from time to time. An overflow to the bilge, rising from the bottom of the well to

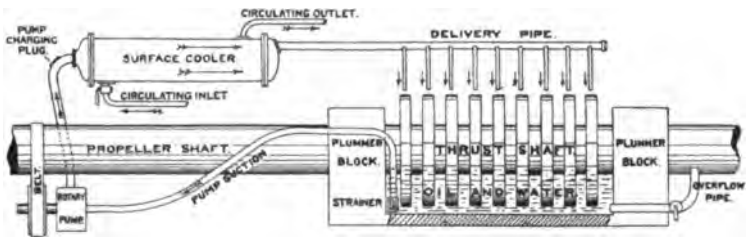


FIG. 158.—Cooling Systems for Thrust Blocks.

about 2 inches above the bottom of the shaft between the collars, allows any excess of water to escape without wasting the oil. The cooling arrangement was next fitted, as shown in Fig. 158, and consists of a cooler of the surface condenser type, containing about 20 square feet of cooling surface for each block, with circulating water taken from the ordinary water service and discharging into the bilge. The mixed mineral oil and fresh water is sucked from the oil well by a rotary pump, driven by the propeller shaft and a leather belt, and discharged through the cooler and a separate

delivery pipe, about $\frac{3}{8}$-inch bore, to each collar. With this arrangement no difficulty has been experienced in keeping the thrust blocks cool when running at full power; the temperature of the lubricant ranges from 100° to 114° F., and the wear of the collars during a year's working is inappreciable. Experience shows that it is better to trust to the centrifugal grooves only and to plug the ordinary oil pipes when under way; also that olive oil is preferable as the lubricant when running at full power, and that a rotary pump (cog-wheel type) is much more efficient for the purposes of circulating the mixture than a pump of the centrifugal type. There is undoubtedly an improved efficiency—probably about 2 per cent, or 600 I.H.P., saved at full power,—and the quantity of oil used is decreased.

Stern Tube.—The engine is connected by the shafting with the propeller, which is outside the ship, and where the shafting passes through the hull, it is necessary that it should be not only free to rotate the propeller, but also that the water should be excluded from entering the hull. The propeller is fitted well clear of the hull at its largest transverse section, to allow a free flow of water to it in a manner as little disturbed as possible; and there is consequently a great length of shafting abaft the aftermost plummer block which requires supporting. This support is given by the stern-tube bearings and a bracket, which is fitted just in front of each screw in twin- or multiple-screw ships.

Inside the stern bearings, and for rather more than the whole length of the stern tube, the shaft is sheathed with a gun-metal casing, which forms its bearing or rubbing surface, *F*. The sheathing is either forced or shrunk on to the shaft, which it fits closely in the vicinity of the bearings. In many instances this casing is undercut both internally and externally between the bearings. The space between the sheathing and the shaft is filled with a mixture of red lead and linseed oil, forced in by a pump, and left at a pressure of about 30 lb. per square inch. Care is taken to allow the air to escape while the mixture is being pumped in.

The stern tube itself, which forms the support for the stern bearings, is built as part of the hull, and the ship is specially strengthened and stiffened at this part for the purpose (Fig 159). The inboard end is riveted to the transverse framing, and generally this framing forms a water-tight bulkhead between two compartments of the ship. The bulkhead is stiffened by rings, *C*, and the

tube is riveted to the ring *C*, which is generally an angle steel. The after end is of similar construction, but without the packing gland arrangements, *G* and *E*. At the forward and after ends of the tube, cast-iron rings, *D*, are riveted to the transverse framing. These are accurately bored for the reception of a long gun-metal sleeve, which is made with suitable recesses for the bearing strips at each end, and at the inboard or forward end a stuffing box, *E*, is also formed. The bearing strips are made of *lignum vitæ*, and are dovetailed into the sleeve, which should not be allowed to come in rubbing contact with the sheathed shaft and its revolving surface. In nearly all cases, especially for the outer end, the strips are fitted into semicircular gun-metal casings, in order that they can be easily withdrawn for examination or refit, and then screwed studs are fitted to prevent these casings with their contained strips from being carried round with the revolving shaft. The strips in the lower part of each bearing are generally fitted with the grain end on to the shaft, which considerably reduces the wear.

The water outside the ship is allowed free access to the space between the strips and between the revolving and fixed casings; and a cock is fitted at the inboard end at the top of the stuffing box, to allow water to circulate when required. To prevent the unnecessary admission of water at the stern-shaft bearing, a packing gland, *G*, is fitted and packed with soft packing, generally gasket well soaked in tallow. The nuts on the gland studs are fitted with cogs, which are geared into a single cogged-ring, to facilitate adjustment by screwing up the gland evenly.

Shaft, or "A," Bracket.—In twin-screw ships it is sometimes necessary to support the shaft outside the ship. This is done by means of a bracket, fitted just in front of the propeller. The upper end of the bracket is generally riveted to the protective deck, where one is fitted which projects outboard for the purpose; the lower end is riveted to a shoe plate, projecting from and forming part of the ship's keel. The bracket is fitted with a *lignum vitæ* bearing in a similar manner to that fitted in the stern bearings, and it is made in two semicircular parts, bodily removable, to allow for the removal of the shaft from the stern tube. The space between the boss of the screw and the bracket is cased in as far as possible (Fig. 287), to prevent ropes or other obstructions being caught up by the revolving shaft and its connections. The casing is preferably made of zinc, and it then serves as a protection against galvanic action; the

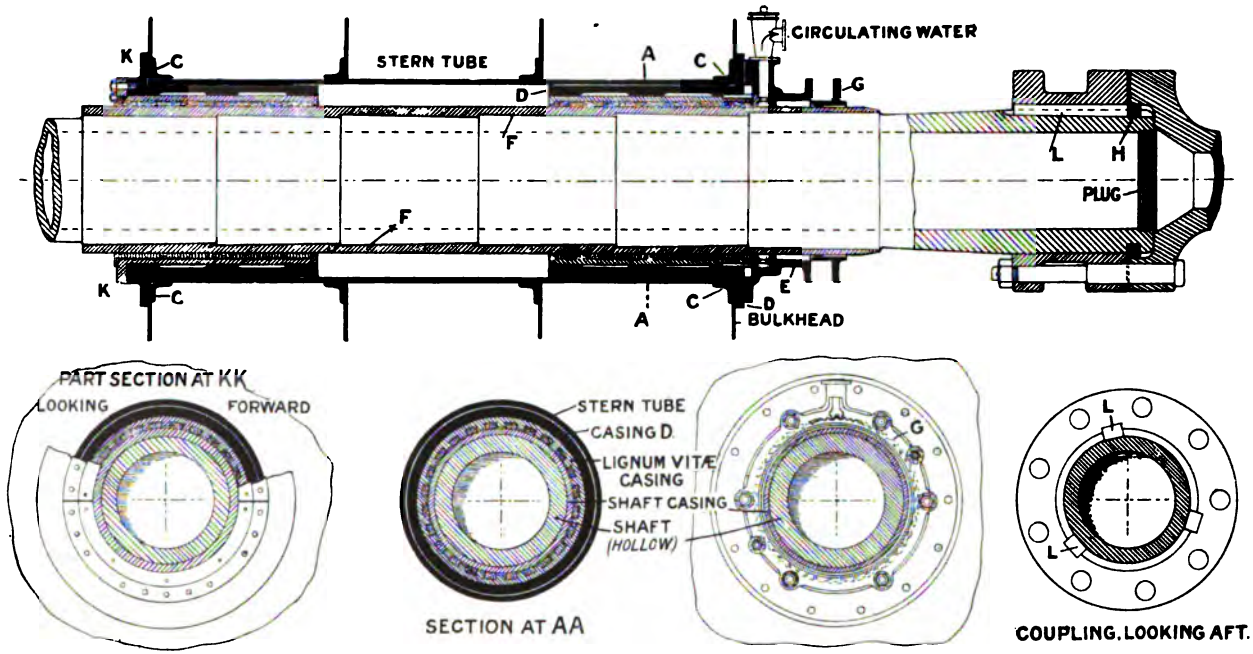


FIG. 159.—Stern Tube, Shaft, and Loose Coupling.

surface of the rounded part of the bracket is also covered with zinc in many ships.

Stern-shaft Coupling, or Loose Coupling.—The aftermost coupling inside the ship is made in such a way that the shafting inside the stern tube can be withdrawn outboard when required; but when the engine is working astern the withdrawal must be prevented.

A simple type of loose coupling is shown in Fig. 159, and is frequently used. The coupling at the after end of the intermediate shaft is suitably made with a hollow space (shown). This coupling is solid with its shaft. The loose coupling is fitted to the forward end of the stern shaft, and is secured to it by three or four keys, *L*, and therefore revolves with it. This coupling is generally made of wrought iron where a steel shaft is employed, as it is then less liable to set fast. The stern shaft projects beyond its coupling, and into a recess two half-rings, *H*, fit accurately, with no fore-and-aft play. The coupling bolts draw the couplings together after the half-rings are inserted.

When going ahead, the thrust is imparted from the stern shaft partly by the ring and partly by a shoulder formed on the shaft against which the coupling is butted; when going astern, the half-rings take the pull on the shaft and transmit it to the thrust block.

When the shaft is hollow, a plug is fitted to the forward end of the stern shaft. In case of fracture, it prevents the entry of water into the ship through the hole in the shafting. In some ships, instead of the half-rings a special capped nut is fitted with a keep to prevent it from slacking back. The thread used on the end of the shaft for the reception of the nut should be of the opposite hand to that of the screw propeller to which the shafting is attached.

Method of Removing the Shafting.—All the shafting inside the ship must be lifted into it and out of it through the hatchways, unless the couplings are separate fittings.

When the shafting outside the ship is of such great length that docking becomes difficult on account of the length of floor required, it is sometimes made in two pieces, joined by an outside coupling just abaft the stern tube and clear of the hull. In big ships the part external to the hull is sometimes encased in a thin-sheet steel tube, which extends as far as the shaft bracket, and is generally

secured to it. The internal diameter of this tube casing must be large enough to clear the outside coupling. To remove the outside shaft, it is necessary to first remove the propeller and boss from the end of the shaft and part of the exterior casing mentioned above, then to support the weight of the shaft independently, then to remove the bearing in the stern bracket and take out the coupling bolts; then by withdrawing the shaft aft a little, the coupling is allowed to fall clear of that on the stern shaft, and the outside shaft can then be drawn forward sufficiently to clear the bracket. After the outer shaft is removed, the stern shaft can be withdrawn outboards from the stern tube. It is necessary to so proportion the lengths of the shafting that any one can be removed or renewed without disturbing any of the permanent fittings. In some cases it is not necessary to first remove the outside length to withdraw the stern shaft, as the former can be taken back a sufficient distance through the bracket bearing.

In single-screw ships sufficient length must be allowed in the dock for the removal of the shafting outboards, and a little consideration shows that this is not very convenient, as it involves a very long dock.

Since about 1900 the outside coupling and the stern-shaft casing have not been fitted in the British Navy. The propeller shafts, although of great length (about 80 feet in the example shown in Fig. 154), can be removed by uncoupling at the loose coupling, removing the propeller, and, after taking the weight of the shaft on slings, removing the semicircular bearing cases in the "A" bracket. Then, by withdrawing the shaft aft until its fore end is clear of the stern tube, the fore end can be moved away (as shown in dotted lines), and the shaft moved forward until its after end is clear of the "A" bracket. The bearing cases and the hole in the bracket are made sufficiently great in diameter to allow the shaft to be moved out of line.

Safety Stops for Bearing-Bolts.—The absolute prevention of the nuts on bearing-bolts from slacking back is of the utmost importance. A slight slackness in a bearing produces "hammering," which gradually increases the slackness by spreading the white metal over the edges of the bearing, and may then produce sufficient momentum, on the reversal of the direction of motion, and distortion on one of the bolts, to fracture it and cause a breakdown. If the bearing is too tightly screwed up, it leads to overheating.

A stop suitable for large engines is shown in Fig. 160. The nut *A* is spigoted into a washer ring, *B*, which is prevented from turning by one or two dowel pins, *C*. After the nut is hardened up, a set screw, *D*, is entered in a groove cut in the spigot, and secures the nut from slacking back. A series of countersunk holes should be made, by the point of a drill, in the groove, so that the case-hardened point of the set screw *D* should always enter one of

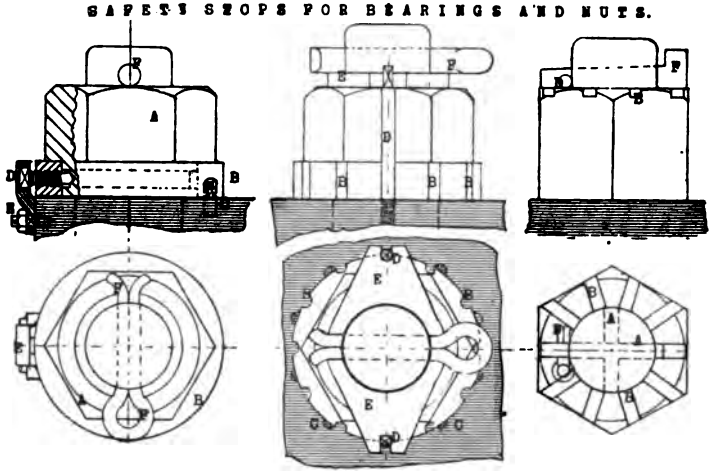


FIG. 160. FIG. 161. FIG. 162.

Safety Stops for Bearing-Bolts and Nuts.

them, as shown. The head of the set screw is prevented from turning by passing a socket over it, which is contained in a bracket, *E*, fixed to the body containing the bearing by square-necked studs, nuts, and split pins. As an additional precaution, the bolts are generally fitted with a substantial split pin, *F*, outside the nut, and with steel washers in any space between it and the nut when hardened up. For convenience in adjustment, the corners of the nuts should be numbered and a radial dial cut into the face of the washer, so that the settings can be recorded and checked when making a fresh adjustment.

Another method used with fast-running engines is shown in Fig. 161. The washer forms part of the nut, and across its edge a number of semicircular grooves, *B*, are cut, generally twelve. Below the face of the washer, and in line with these grooves, holes,

C, are drilled and tapped, generally ten; and two long studs, *D*, can be fitted, passing through two grooves and into two holes; these studs fix the nut in the required position. The studs are made with squares on them, which are held from turning by a plate, *E*, fitted under the split pin *F* and across the end of the bolt. If the screw thread on the bolt is **four** to the inch (a usual number), and the nut can be set in thirty positions, agreeing with the number of holes given above, then it is possible to make an adjustment of $\frac{1}{120}$-inch on the bearing.

The method shown in Fig. 162 is generally used for piston-rod nuts. A taper key, or cotter, *F*, is fitted in the top end of the bolt and partially sunk into the nut. By cutting two keyways, *A*, in the end of the bolt at right angles, and providing five slots, *B*, in the nut, an adjustment of one-twentieth of a turn of the nut is obtained; if the number of threads to the inch is four, a longitudinal difference of $\frac{1}{80}$-inch is obtainable. The cotter is secured in place by a split pin, *D*.

Size of Shafting.—If hollow shafting is used, and it is generally, it must be made about 3 per cent greater in diameter for the same strength as a solid shaft, allowing the diameter of the hole to be 55 per cent of the external diameter. The saving in weight is then about 26 per cent on the solid shaft for the same strength. For a certain large engine the diameters of the typical parts of the shafting are as below:—

	inches.
Crank pin	23·25
Crank shaft	21·05
The crank shaft has to take a large bending moment as well as torsional stress.	
Thrust shaft	20·1
This is most difficult to repair, and between the collars it is in addition made $\frac{1}{4}$-inch greater in diameter than at the outer ends.	
Intermediate shaft	19·5
This is most easy to repair and made smallest in diameter, but as it is generally well supported, is nearly as strong as the remainder.	
Stern shaft, inside tube	21·0
This is difficult to repair, almost inaccessible.	
Stern shaft, outside ship	23·0
Frequently of great unsupported length, and therefore subject to a bending moment due to its own weight and also to the working of the ship; also subject to corrosion from sea-water and galvanic action.	

Pressure allowed on the Rubbing Surfaces of Bearings.—There is an enormous difference in the pressures allowed per square inch of rubbing surface in the various bearings, even when the lubrication is nearly as efficient as it can be made; and, as an object-lesson in proportion, some typical figures are given below, with some of the principal reasons for their existence.

For bearings, such as the cross-head, crank-head, and shaft bearings, the effective area on which the pressure is exerted is considered to be that of the "diameter of the pin multiplied by its length," and a little consideration shows that this is very nearly the correct theoretical amount.

Maximum pressures allowed per square inch of effective surface:—

Connecting rod, top-end bearing . . .	1200 to 1250 lb.
There is only an oscillating motion on this bearing, not a complete rotation, with a very low velocity of rubbing.	
Connecting rod, crank-end bearing . . .	500 „ 550 „
There is a complete rotation, one revolution to each revolution of the engine, but with a varying velocity.	
Guide surface for **ahead** working . . .	75 „ 140 „
This small amount is due to difficulty in retaining the lubricant on the rubbing surfaces.	
Main bearings	300 „ 350 „
Thrust rings, effective surface only . .	40 „ 70 „
This small amount is due to difficulty in retaining the lubricant, which is easily thrown off by the centrifugal action.	

Pressure on the Guide.—In considering the motion of the ordinary reciprocating engine, the various forces and resistances acting in various directions must also be included. In Fig. 163 a diagram is given of the forces with their directions. The full lines show any position of the crank *AC*, and the connecting rod *BC*, on the down stroke. Suppose the total force, or pressure, exerted on the piston to be P, then this force P acts in a line, B′B. At B part of this force is transferred to the connecting rod, and to alter the direction of this force into the direction of the connecting rod for transference to the crank, a variable force, or resistance, G, is necessary. G is the resistance of the guide, and acts in the direction shown by the arrow, that is, perpendicularly to the guide surface.

At the point B we have three forces, which may be momentarily considered in equilibrium. Denote the angle CBA by θ. Then—

$$G = P \cdot \tan \theta,$$

and at the top and bottom of the stroke (*i.e.* the dead points) the angle vanishes; consequently—

$$\tan \theta = \text{zero}.$$

Therefore G = zero at these points, and G will be a *maximum* when tan θ is a maximum; that is, when θ is a maximum. A little consideration shows that θ is greatest when the angle BAC is a right angle.

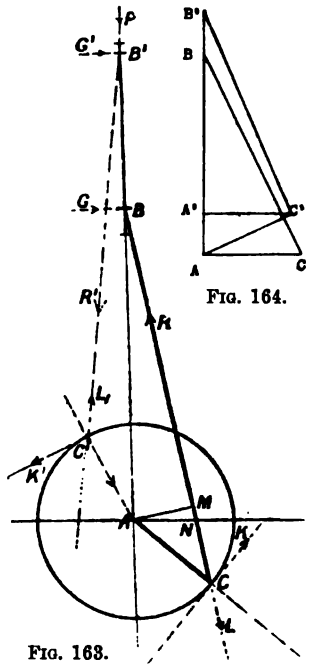


Fig. 163. Fig. 164.

Pressure on Guide.

Proof (Fig. 164).—In the triangles *BAC* and *B'A'C'*, *A'C'* is drawn perpendicular to *AB*. *BC* is equal to *B'C'*, the length of the connecting rod; *AC'* is equal to *AC*, the length of the crank; and each is greater than *A'C'*.

Therefore the angle *ABC*, which is subtended by *AC*, is greater than the angle *AB'C'*, which is subtended by *A'C'*. *Q.E.D.*

Area of Guide Surface.—The area of rubbing surface on the slipper on the crosshead at the end of the piston rod must be made of a suitable area for the maximum pressure on the guide, or G. For fast-running engines, an area which limits the maximum pressure when going "ahead" to about 40 lb. per square inch of area is allowed.

Turning Moment or Crank Effort.—Let the crank, when in any position, AC, make an angle CAB equal to x with the vertical line, and let AC equal b, and BC equal a; draw AM perpendicular to BC. Then if P be acting downwards at B—

$$\text{The turning moment at C} = R \times AM$$
$$= R \times b \times \sin(\theta + x).$$

$$\text{But R, the force acting on the crank at the point C} = \frac{P}{\cos \theta}.$$

$$\text{Therefore the turning moment} = \frac{P \cdot b \cdot \sin(\theta + x)}{\cos \theta};$$

and as b and a can be measured, the angles θ and x can be calculated.

Adopting a factor of safety for torsion of not less than 6, and for tension of about 8, the diameter of the crank shaft is given by the formula—

$$D = F\sqrt[3]{\frac{I.H.P.}{N}};$$

and the other diameters are in proportion, where—

D = diameter in inches;
N = number of revolutions of the engine per minute; and
F = 4 to $4\frac{1}{2}$, according to the quality of the material.

CHAPTER XVI

THE SLIDE VALVE AND ITS MOTION

Necessity of Slide Valve.—In Chapter I. it has already been explained how, by the addition of heat to water and the consequent formation of steam, the piston can be moved. In the ordinary reciprocating engine the piston is connected by means of the piston rod and connecting rod with a crank arm on a shaft; thus the piston moves only a limited distance each way from its middle position. The total distance which the piston moves is called the stroke, or travel, of the piston, and is equal to twice the *throw of the crank*; the throw of the crank is the distance between the centre of the crank pin and the centre of the shaft on which it is fitted. It is necessary to have an automatic arrangement by which the pressure can be admitted to each side of the piston alternately, and at the same time allow the exhaust steam to escape from, and thus decrease the resistance on, the opposite side of the piston; this is effected by the slide valve. From its peculiar position as the distributor of the steam, the slide valve is a most important element of the ordinary steam engine, as by its means all the processes of admission, cut-off, expansion, release or exhaust, and compression are produced by its proper motion.

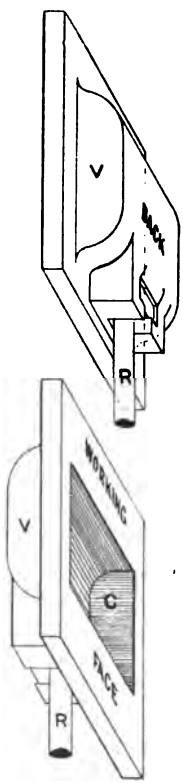


FIG. 165.—Common Slide Valve.

Common Slide Valve. — Fig. 165 shows a perspective view, front and back, of an ordinary D, or locomotive slide valve; Fig. 165A shows the cylinder, flat face on, which the valve moves; and Fig. 165B

shows a section through the cylinder face with the slide valve in the middle of its stroke. V is the valve which is moved up and down on the face of the cylinder AB by the slide rod R; the method of obtaining the motion of the valve will be considered later. At A is a steam port, which is in connection with the upper

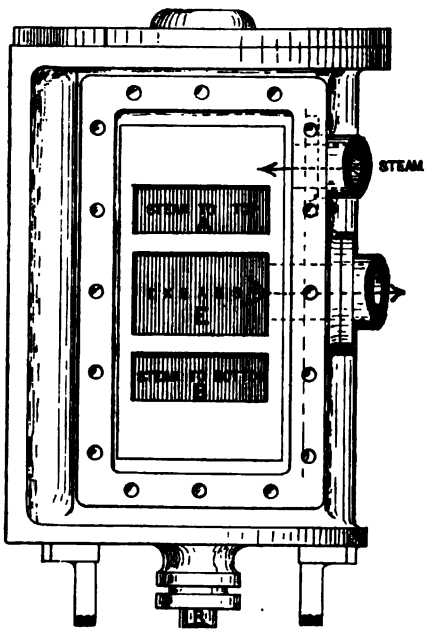


FIG. 165A.—Cylinder Face.

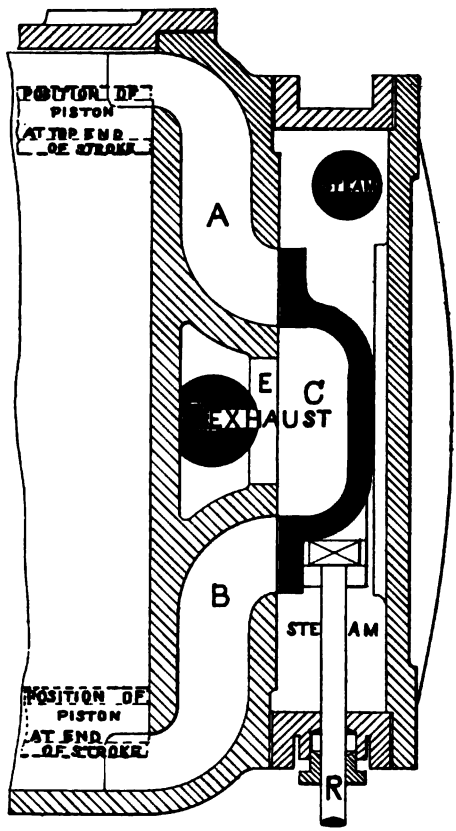


FIG. 165B.—Section through Valve and Cylinder Face.

part of the cylinder above the piston; at B is another port, in connection with the lower part of the cylinder below the piston; and at E is another port, of about twice the vertical width of the other cylinder ports, A and B, which is connected by a pipe with the exhaust. Opposite to E is a large cavity, C, in the slide valve, known as the exhaust cavity; the travel of the valve is such that C and E are always in open connection.

Steam is admitted to the engine through the regulating valve, and the space outside the valve, away from the cylinder, is filled with steam under pressure; the pressure keeps the valve pressed up against the cylinder face. Now, if the valve be moved downwards, the cylinder port *A* is gradually opened, and steam is admitted above the piston, pressing it downwards; at the same time the port *B* is opened and connected through *C* with the exhaust at *E*. The difference of pressure above and below the piston produces a motion downwards. Similarly, if the valve is moved upwards, steam is admitted below the piston, the port *A* is placed in connection with the exhaust, and the difference of pressure produces motion upwards; and so on, alternately, in accordance with the distribution of pressure by the valve.

The operations performed by the valve are considered later, when the means of producing its motion have been more clearly explained.

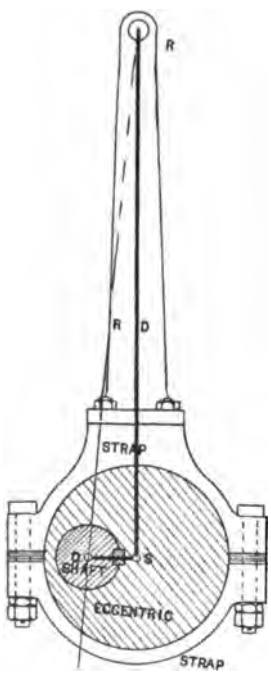


FIG. 166.—Eccentric and Rod.

Production of Motion of the Valve.—The most usual method of moving the valve is from an eccentric fitted on the shaft. The piston reciprocates and produces a revolving motion of the shaft through the medium of the crank. If another crank is fitted on the shaft, it can be made to work a reciprocating piece, such as the slide valve. This is done in ordinary practice; but an eccentric is substituted for the crank, as it is not convenient to cut the shaft for the purpose.

In Fig. 166 an eccentric is shown keyed to the shaft, and embraced by the eccentric strap; the upper half of the strap is secured to the eccentric rod, which at its top end *R* is connected with the slide rod, either directly or through the link. The eccentric revolves with the shaft, inside the strap, and produces a reciprocating motion of the eccentric rod and slide rod.

If the crank pin of an ordinary crank shaft be enlarged sufficiently to include the shaft within it, the pin becomes an eccentric. The eccentric is therefore a crank, *OS*, fitted in a special way. The distance between

the centre line of the shaft and the centre line of the eccentric *OS* is equal to the throw of the eccentric or virtual crank, and the travel of the valve becomes equal to twice the length of the virtual crank, or *radius of eccentricity*, as it is frequently called.

Definition.—The *radius of eccentricity* is the length of the virtual crank, and therefore *one-half the travel* or stroke of the slide valve; it is the distance between the centres of the shaft and the eccentric.

Normal Slide Valve.—The early slide valves were made without lap—that is to say, when the valve was in the middle of its stroke, any motion of the valve opened the cylinder to steam, either above or below the piston. In Fig. 165B a normal slide valve is shown in the middle of its stroke, and with no part of the valve overlapping the cylinder ports on either the steam or exhaust edges. For working the normal slide valve the eccentric radius is fixed at right angles to the crank, so that when the piston is at either the top or bottom of its stroke the valve is near the middle position, and just covers both the cylinder ports.

In Fig. 170 the crank is represented by *OP*, and the eccentric radius by *OS*; in the first position a movement of the crank in a clockwise direction pushes the valve upwards, admitting steam below the piston, and opening the other port to exhaust; therefore the piston will promote rotation in the clockwise direction. The cylinder is open to steam from below during the upward stroke of the piston, and the upper side of the piston is open to exhaust. Both ports are closed near the top of the stroke, and the crank continuing its rotation, from its own momentum or some external cause, steam is next admitted above the piston, and the exhaust is opened to the bottom of the cylinder. The motion produced is therefore in a clockwise direction.

If an eccentric is fixed to the opposite side of the crank, as shown by the dotted lines *OS'*, a movement of the crank in a clockwise direction pulls the valve downwards, admitting steam above the piston, which tends to return the latter to its original position. But if the crank is moved anti-clockwise, the motion of the valve is such that the steam is admitted and exhaust opened to assist motion in the same direction.

From the foregoing it may be concluded *that the eccentric leads the crank in the direction of rotation.* This is true, however, only when the steam enters the cylinder from the outer edges of the valve (as in Figs. 165 and 167), and generally for all flat valves.

The cylindrical valve shown in Fig. 181 is an example of an exception.

Lap.—The normal valve admits steam for the whole of the stroke of the piston. It is economical, however, to arrange for the steam to be used expansively by cutting off the supply at some point before the end of the stroke. This is effected by making the valve overlap the steam ports of the cylinder, as in Fig. 167. If lap is fitted as in Fig. 167, and the eccentric radius is set at right angles to the crank, it is evident that the crank must be moved a certain distance before steam is admitted to the cylinder. Although the cut-off now takes place earlier, the steam admission takes place later, and the amount of opening to steam has been decreased.

To obtain the same maximum opening to steam, it is necessary to increase the travel of the valve, that is, to increase *OS*; but after this is effected the steam still enters the cylinder too late.

By advancing the position of the eccentric radius relative to the crank until the valve begins to open at the beginning of the stroke of the piston, and following up the operations of the valve with respect to the piston, it will be noticed that all the operations are performed as required, viz. admission, cut-off, expansion, and release.

Definition.—The lap of the valve is the amount the valve overlaps the cylinder steam port when it is in the middle of its stroke; the amount of overlap on the steam edge is called the *steam lap*, and the amount it overlaps the exhaust edge is called the *exhaust lap*.

The addition of steam lap to the early locomotive engines produced an economy of fuel of 25 to 50 per cent.

Exhaust lap is very small in compound expansion engines, and is frequently *negative* (Fig. 167).

Lead. *Definition.*—The amount of opening of the port to steam when the piston commences its stroke is called the lead of the slide valve.

When steam is admitted at the beginning of the stroke, it takes an appreciable time to fill the cylinder, as the admission is gradual; therefore the full pressure does not act on the piston at once. At the end of the stroke the reversal of direction of motion of the piston produces great shock, unless some means are taken to cushion it just previous to the reversal. By advancing the eccentric a small amount (Fig. 167A), the slide valve is given sufficient lead to admit steam before the end of the stroke. This forms an elastic

LAP OF SLIDE VALVES.

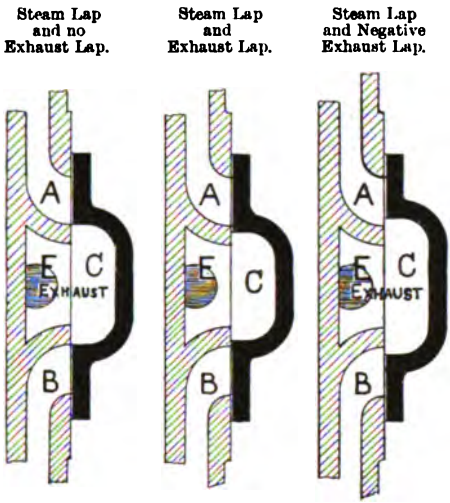
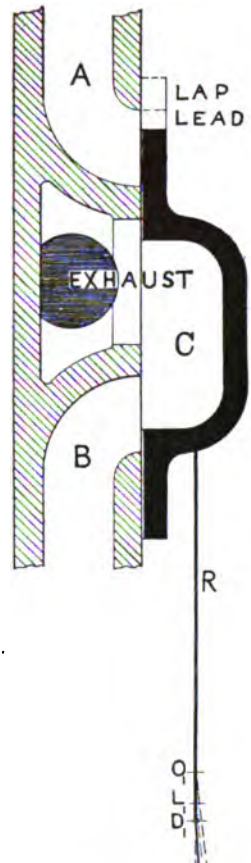


FIG. 167.—Lap of Slide Valve.



$OL = O_1L_1$ = Steam LAP.
$LD = L_1D_1$ = Steam LEAD.
Angle KOS = Angle of Advance of Eccentric due to LAP + LEAD.

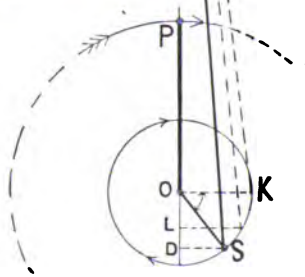


FIG. 167A.—Angular Advance of the Eccentric.

cushion, on which the piston is brought gently to rest, and at the same time assists the reversal of its motion. The increased advance of the eccentric used to obtain the lead makes all the operations performed by the valve earlier.

Angle of Advance of the Eccentric.—The total angle the eccentric is advanced for the necessary lap and lead is called the angular advance.

The angle varies for different types of valves and engines, but is generally about 40°. By reference to Fig. 167A, a little consideration makes it evident that when the eccentric radius *OS* is at right angles to the line of travel of the valve, the valve is not quite in the middle of its stroke, but rather closer to the shaft; this is due to the obliquity of the eccentric rod. The length of the eccentric rod is very long compared to that of the eccentric radius, and consequently the obliquity is very small; and by neglecting obliquity the following construction gives the angle of advance. In Fig. 167A, *OP* represents the position of the crank. *OL* is made equal to the lap, and *LD* is made equal to the lead. Make *OS* equal to the radius of eccentricity or half-travel of the valve, and describe a circle with *OS* as radius; draw *DS* perpendicular to *OD* and cutting the circle at *S*; then *OS* is the position of the eccentric radius or arm, and the angle *KOS* is the angle of advance

Definition.—The angle of advance may also be defined as the angle at which the eccentric radius stands in advance of the position that would bring the valve to its mid-stroke, when the crank is on the dead point (*i.e.* in line with *OR*).

Width of Exhaust Cavity.—So far, only the steam edges of the valve have been considered. These govern the admission and cut-off of the steam, while its compression and release are governed by the exhaust edges. It is arranged that the exhaust is as free and unrestricted as possible, but it is necessary that the exhaust should be closed before admission commences. The decreased pressure, obtained by the expansion of the steam in the cylinder, produces a large increase in volume, which has to pass through the port in equal time, *i.e.* at each stroke; consequently the width of the cylinder ports, through which the exhaust has also to pass, is about 50 per cent greater than the maximum amount that the valve uncovers of the ports at the steam edges; but the full width of each port is uncovered by the exhaust edge of the valve.

The width of the exhaust cavity in the valve must be such as

to comply with the above conditions, and also to suit the width of the exhaust port fitted in the cylinder false face.

Operations of the Slide Valve.—In Fig. 168 a series of positions of the crank, the piston, and the slide valve are shown relative to each other. For clearness the eccentric radius is shown

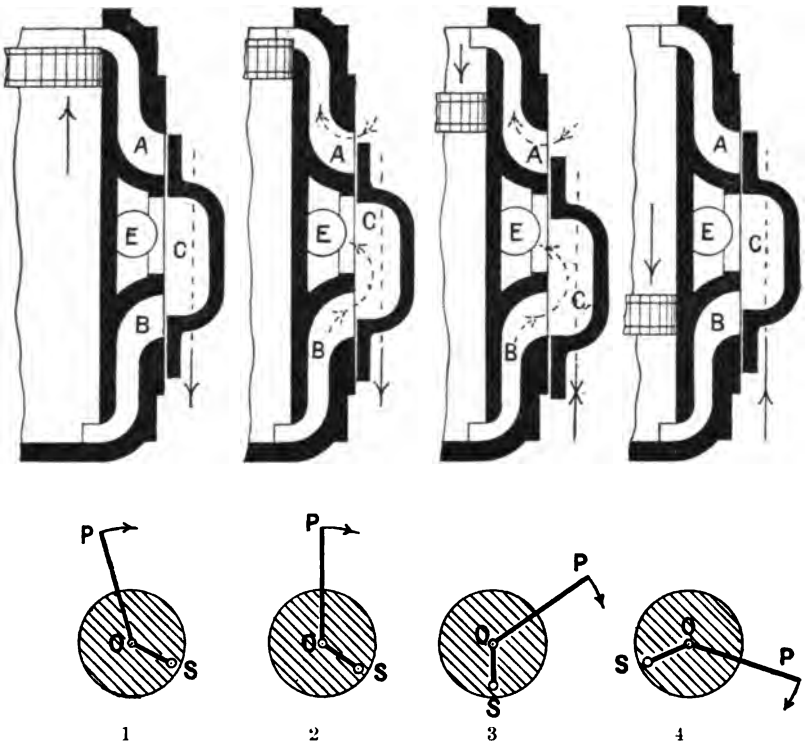


FIG. 168.—Operations of the Slide Valve.

1. Admission.
2. Lead, Down-stroke.
3. Maximum opening to steam.
4. Cut-off, Down-stroke.

as a crank arm turning with the main shaft. *OP* represents the crank, *OS* represents the eccentric arm, and the excess of the angle *POS* over a right angle is the angle of advance.

Admission.—When moving in the direction of the arrow, the piston and crank are moving upwards. Just before the end of the upward stroke of the piston, the valve moving downwards uncovers the upper cylinder port and admits steam above the piston; the instant at which the port begins to open is called the instant of

admission. Owing to the lead, the piston has not quite reached the top of its stroke when admission begins.

Cut-Off.—When the crank reaches the position shown in figure (4), the piston has travelled some distance downwards while the valve has travelled downwards, and, returning upwards, cuts off the

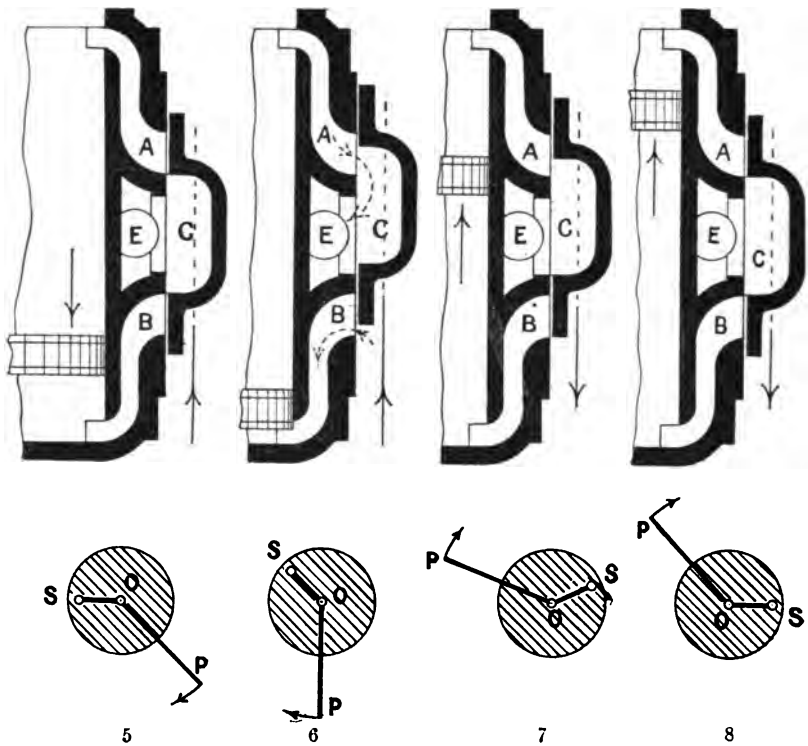


FIG. 168A.—Operations of the Slide Valve.

5. Release, Down-stroke.
6. Lead, Up-stroke.
7. Cut-off, Up-stroke.
8. Compression, Up-stroke.

admission of steam; the steam expands as the piston continues its stroke until release takes place.

Release.—The motion of the valve upwards next uncovers the upper cylinder port connecting it with the exhaust, and the instant at which the port begins to open is called the instant of release; the steam escapes from the cylinder into the exhaust cavity through the upper cylinder port.

If there be zero exhaust lap, the instants of release and compression (which latter is next described) are coincident; if there be positive exhaust lap, compression begins before release; but if there be *negative* exhaust lap, which is more usual, release begins before compression.

Compression.—At this instant no steam can pass into the cylinder, as both ports are closed, and no exhaust can take place for the same reason. The bottom port, *B*, has just closed connection with the exhaust; the steam remaining below the piston is imprisoned, and compression begins. If the link, at this point of the stroke of the piston, is drawn over towards its position for astern working, it will be noticed that compression would have begun earlier; *i.e.* by "linking-up" or "shortening the link," compression begins earlier.

Readmission.—At a point similarly situated to that of admission, but on the down-stroke, compression begins below the piston just previous to the instant of readmission, *i.e.* admission for the up-stroke. At the instant of admission for the up-stroke the crank has advanced to nearly the bottom dead centre, the steam above the piston has been released by the valve opening to exhaust through the cylinder port *A*, while the steam below the piston has been compressed by the valve covering the exhaust through *B*, and *B* is just at the instant of admitting steam.

On the up-stroke, the processes of cut-off, release, compression, and readmission for the down-stroke are repeated; and so on in continuous rotation.

Eccentric, Strap, and Rod.—The details of the construction of an eccentric and strap for a large engine are shown in Fig. 169. The eccentric, or sheave, is sometimes made in one piece and slipped over the end of the shaft and keyed to it; but this is only possible in a few instances. Generally it is made in two parts, *A* and *B*, tightly embracing the shaft *C*, and secured together by two bolts, *D*. The two parts are filleted at the butting faces, which are machined up to true surfaces. The smaller part, *B*, is generally made of forged steel, the greater strength of which allows the outer part of the sheave to be brought more closely to the centre of the shaft, and thus reduces the diameter of the eccentric. The larger part, *A*, is generally made of cast iron, lightened by holes, and keyed to the shaft by one or more keys. The key should be carefully fitted, and made a driving fit on its sides and only a moderate fit radially,

so that the part *A*, as well as *B*, should be in firm contact with the shaft throughout its circumference. Two set screws, *E*, with check nuts, are used for temporarily fixing the eccentric in position on the shaft when setting the slide valve, before the actual keyway is cut. They are also used to prevent the sheave creeping along the shaft, and, if the fit is faulty, to take up any slackness between the shaft and eccentric.

The right-hand part of Fig. 169 shows a pair of eccentrics and straps: the wide one, which is in section, is used for ahead working; and the other, shown in side elevation, is used for astern working. The sheaves fit over an enlarged part of the shaft, so that the cutting of the necessary keyways does not weaken it.

The *eccentric strap* is made in two parts, generally of steel, but frequently of gun metal, and in either case is lined with white metal, which forms the rubbing surface with the sheave. The parts are made with an inwardly projecting flange on each side, to prevent the oil being thrown out too easily by centrifugal action, and these flanges also maintain alignment between the eccentric and slide rod. Gun-metal liners are fitted between the butting ends of the upper and lower parts of the strap: the thicker liners form distance pieces, and do not touch the sheave; the thin liners are used for correcting adjustment. When the engine is at work the straps tend to contract at the lips, especially if the bolts are not sufficiently tightened up on the liners, and to produce sufficient stiffness the straps are sometimes made with ribs.

The *eccentric rod* is made of steel, with a foot, and a spigot below the foot, which fits into a recess in the top part of the strap, and maintains alignment. Two studs, generally with collars and nuts, are used to secure the eccentric rod to the shaft.

The *bolts* securing together the two parts of the sheave and the two parts of the strap are made of steel, with the usual stretching lengths, nuts and washers, and split pins, and a socket key under the head of the bolt, to prevent its turning in the hole. Cotters were sometimes used instead of nuts, but they are not so satisfactory, principally because the butting surface is not large enough.

Setting the Slide Valve.—The eccentric is first secured temporarily in place at the supposed correct angle of advance, and where reversing motion is fitted the astern eccentric and its connections are also temporarily secured in their proper places. The crank is then placed first on the top centre and then on the

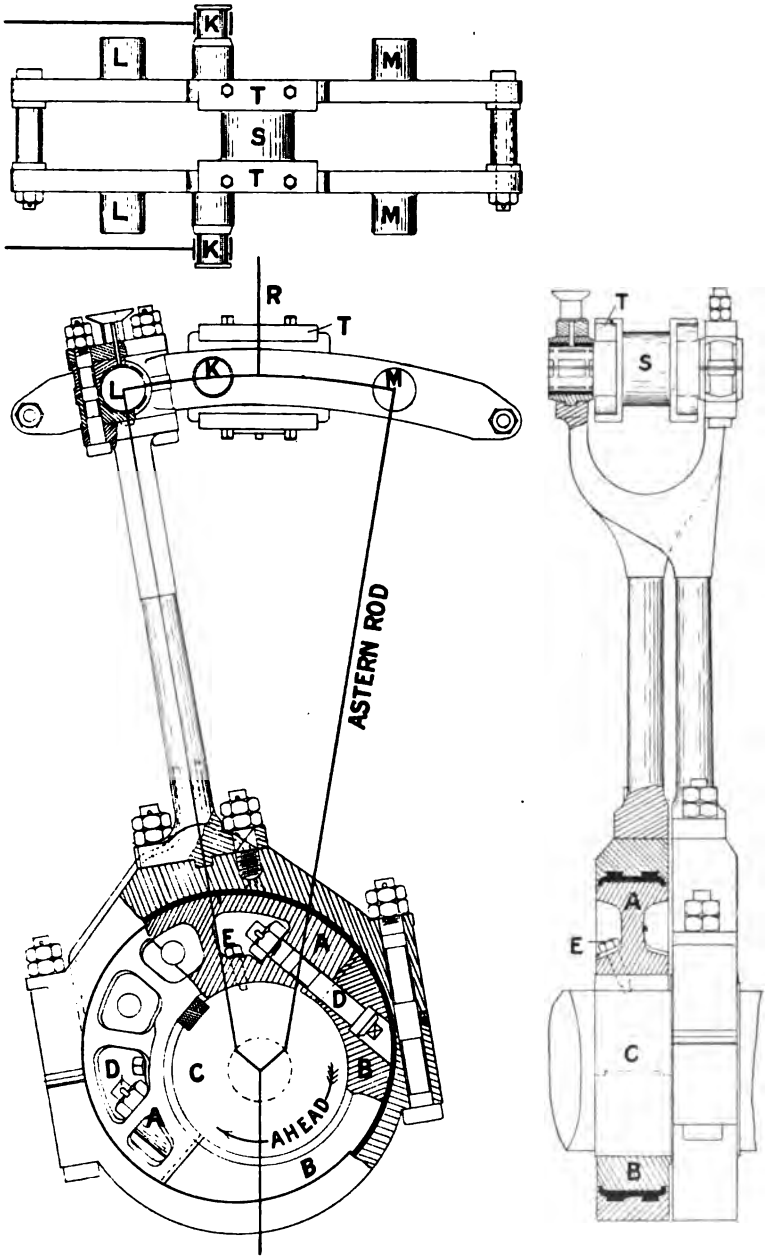


FIG. 169.—Eccentric, Strap, and Rod. with Link and Link Block.

bottom. This should produce an equal travel of the valve above and below the middle of the exhaust port. If incorrect, the position of the valve on the rod is altered to correct the difference. With vertical engines a slightly increased lead is allowed at the bottom to counteract the momentum of the descending weights of the piston rod and connections; this difference is made by altering the position of the valve on the rod *after the angle of advance is corrected.*

The position of the piston on both the up and down strokes is carefully tested with respect to the valve when in its position for admission, cut-off, release, and compression; and if all the adjustments or readings of the valve are correct, the keyways are cut and the eccentrics properly secured in position. When reversing links are fitted, the astern rod and connections should be correctly set before the ahead eccentric is fixed.

Reversing the Engine.—The direction of rotation of the engine can be reversed by fitting two eccentrics on the shaft at the relative positions *OS* and *OS'* indicated in Fig. 170, and by joining the ends of the two eccentric rods by a link, *RR'*, which is attached to the end of the slide rod. By means of bridle rods the link can be drawn over and held in a position such that the rod S'R' is nearly in line with the valve rod, when the action of the eccentric arm *OS* has little effect on the motion of the slide rod, which is now derived principally from that of *R'S'*, shown in dotted lines with altered position of valve.

The act of drawing over the link alters the position of the slide valve on the cylinder face. Steam may be admitted below the piston through the port *B*, and the motion imparted by the steam pressure, therefore, the reverse of the original motion. A careful study of the positions of the slide valve, as shown by the dotted lines, shows that in each case the direction of rotation is reversed by the pressure of steam through the port openings.

The Link.—There are three varieties of link, and room must be given in all of them for the oscillation of the block to which the slide rod is attached. For the purposes of reversing only, the link could in all cases be made straight; but in practice it is generally curved, so that it can be readily set in any intermediate position.

For a perfect linking-up the curvature should be parabolic; but for practical purposes it is part of a circle, whose radius is equal to the length of the eccentric rod measured from the centre of the link, *i.e.* the centre of the pin joint in the end of the slide rod.

The *slot link* is still used for small engines, and was also frequently fitted in horizontal engines. The connections of the eccentric rods are outside the slot; and in a vertical engine they are below it, and nearer the eccentrics.

The *double-bar link* is the most frequently used for marine

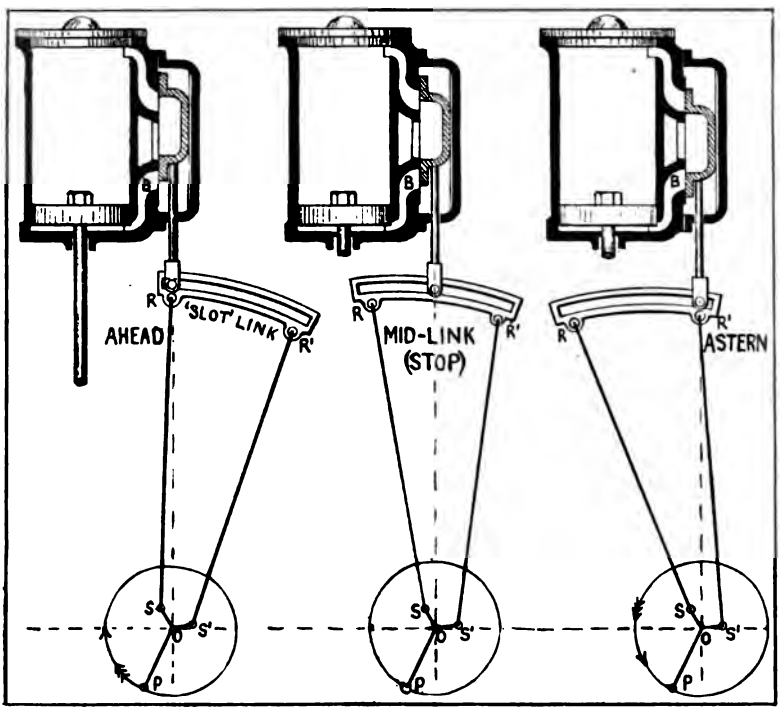


FIG. 170.—Reversing with Two Eccentrics.

engines. It is shown in Fig. 169. Two curved bars are joined together at the ends by bolts, fitted or made with distance pieces. The block *T* slides along the bars between these distance pieces. Each of the two pairs of pins, *L* and *M*, fitted on the outer side of the bars is connected with one of the eccentric rods, which are forked at the top end, for the purpose of fitting twin bearings. These forks are right- and left-handed, rising from the eccentrics, which are fitted side by side on the crank shaft. The *bridle rods* are attached to the central (or other) pins *K* at their outer ends inside the collar shown, and by their means the link can be moved

to or fro to bring the required eccentric rod in line with the slide rod *R*, through the medium of the link block *T*.

The **link block** travels inside the double bars of the link, and is generally provided with a white-metal rubbing surface where it is in contact with the upper and lower edges of the bars. Its central part is provided with a pin, *S*, which is attached to a bearing in the end of the slide rod. The link block is adjustable, and the white-metal lining is generally contained in brasses, secured to the steel block by screws, as shown.

White metal is now generally used for all the working surfaces of the bearings, for the pin joints, *L* and *M*, of the link and weigh-shaft arms *K* and their connections; but the bearings of the weigh shaft itself are of gun metal.

The *solid-bar link* is sometimes used. The link is one solid bar, and can slide between a pair of sector blocks, which are fitted to, and can oscillate in, the end of the valve rod. This arrangement necessitates fitting the connection with the eccentric rods at the extremities of the link, and therefore neither eccentric rod can be brought even approximately in line with the slide rod. The link is consequently "shortened" or "linked up," even when in "full gear," and to obtain the necessary extreme travel of the valve, a greater throw and larger eccentrics must be fitted than with the two types previously described. The larger eccentrics tend to increase the frictional resistance; and although the solid-bar link appears to work fairly well in practice, it cannot be made so efficient as the double-bar link.

Linking-up.—If the link block is set at a point *R* midway between the two points *L* and *M*, the motion imparted to the valve rod is partially neutralised, and any steam admitted tends to keep the piston in the middle of its stroke or thereabouts, and does not allow of a complete revolution of the crank.

If the link be set at some point, say, halfway between *L* and the mid-position, the motion imparted to the valve is derived from both ahead and astern eccentrics, but principally from *L*, and the distance travelled by the valve is considerably less. If the linking-up, either on the ahead or astern side, be not too great, and all the necessary functions of the valve be performed, motion is imparted by the pressure through the openings as before, and the direction of rotation depends on which of the rods, ahead or astern, predominates on the travel of the valve. (See also Fig. 170.)

When linked up, all the operations of the valve are earlier—namely, admission, cut-off (producing greater expansion), release, and compression.

Reversing Gear.—There are three principal methods of reversing the motion of the engine in everyday use:—

1. Changing over the entry and exit of the steam and exhaust, by using a *differential reversing valve.* This system is used for steering and other engines, and is explained in Chapter XXVI.

2. *Valve gear,* either with one or no eccentric, as explained later for Joy's arrangement; the motion is then obtained from the movement of the piston or connecting rod. (See Fig. 187.)

3. Stephenson's **link motion** (Fig. 169).

The most usual method is that adopted by Stephenson, which, although modified in detail, is essentially the same in principle as when first fitted in the *Rocket.* Normal slide valves were used with the original link motion, with practically no lap or lead, and consequently expansion was almost non-existent.

Reversing Gear, All-round Type (Fig. 171).—The link motion is shown in the ahead position, and the crank is at the top of its stroke; the slide valve is of the cylindrical type, taking steam at its mid-length.

The movement of the link to and fro, as necessary for altering the direction of rotation of the engine, is generally obtained in warships by the **all-round** or continuous-motion reversing gear, as shown in Figs. 171 and 172. The notation used in these two figures is similar, but the weigh shaft is differently arranged. In Fig. 172 the crank is at the bottom of its stroke, and the link is central. The gear may be worked either by hand or by steam, and the reversing engine need only be capable of moving in one direction; but frequently a differential valve is fitted with a two-cylinder engine, and the motion is then reversible.

In Fig. 171 the shaft AC is revolved either by the hand-wheel A or by the reversing engine acting on the crank B. A worm, C, is attached to the shaft, and operates a worm wheel, D, on which a pin, E, is fixed. A connecting rod EF works an arm FG, keyed to the weigh shaft G. Another arm, HG, is also keyed to the weigh shaft, with the end H connected by a rod (or rods) HK, with the link LM, at K. R is the centre of the crank shaft; RL and RM are the eccentric rods. According to the direction of motion of the reversing engine, E is made to revolve in a circle, and

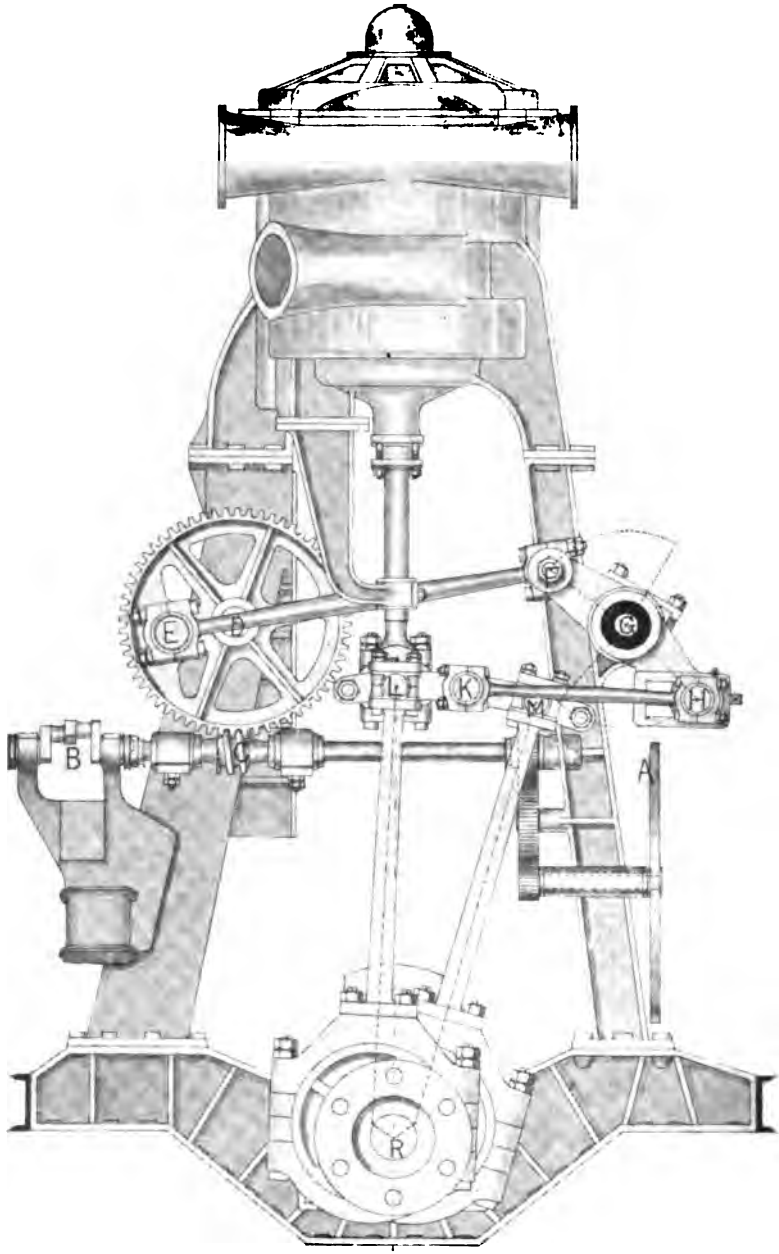


FIG. 171.—All-round Reversing Gear.

moves *F* from right to left or left to right. In this way the link is dragged to and fro, to bring the rod *RM* or *RL* in line with the slide rod, and thus give the desired position for either ahead or astern rotation of the main engine.

REFERENCES TO FIG. 171.

A, Hand-reversing wheel.
B, Cranks of reversing engine.
C, Worm on reversing shaft, gearing into worm wheel.
D, Centre of worm wheel.
E, Pin on worm wheel, making *DE* a crank.
EF, Connecting rod to reversing lever.
F, Pin on reversing lever.
G, Centre of weigh shaft.
GH, Reversing arm keyed to weigh shaft.
H, Twin pins of the *separate link*, which is incorporated in reversing arm.
HK, Suspension or bridle rods.
K, Pins on link to which the bridle rods are attached.
L, Pins on link to which **ahead** eccentric rod is attached.
M, Pins on link to which **astern** eccentric rod is attached.
LKM, **Double-bar link** (Fig. 169).
R, Centre of crank shaft.

This motion is very quick in its action, and requires very careful balancing of the reciprocating parts of the engine, so that it can be stopped in any required position of the link. A double-cylinder reversing engine with a differential reversing motion is preferable for use with all types of main engines. With a continuous-motion gear the distance through which the links are moved is governed by the travel of the pin *E* of the wheel *D*, and there is no chance of accident from jambing in any position. When the movement of the link is derived from a direct motion by a piston or screw gearing, any slight derangement of the automatic cutting-off of the steam to the reversing engine may result in the link motion being jambed in an extreme position, and produce a breakdown.

The "all-round" arrangement is very convenient when waiting orders or raising steam; the reversing engine can be kept continuously in motion, and thus allows the steam to uniformly warm

and free the main engines from water. The slight movement of the main engines when the reversing gear is in motion is not objectionable, and frequently produces a slight vacuum in the

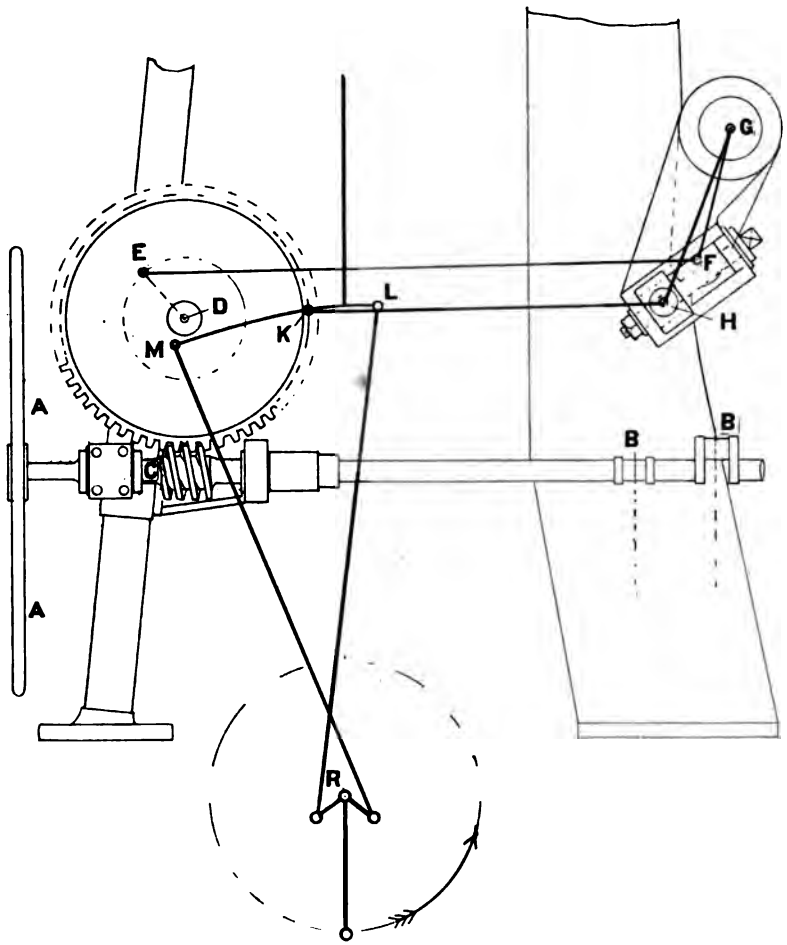


FIG. 172.—All-round Reversing Gear.

condenser if the air-pump is attached to the main engines. A convenient and expeditious means of warming the main engines is almost a necessity with water-tube boilers, where the time required for raising steam is governed by the time required for properly warming the engines.

In Fig. 171 the general arrangement of the cylinder and supports is the same as in Fig. 127. The extreme positions of the reversing arm and lever, the centre lines of the eccentric rods, and the eccentrics, are dotted in. The link block is behind *L*, and is only partly visible; just above the block a circular guide is fitted for the slide rod in the end of the bracket shown.

In both Figs. 171 and 172 the eccentric rods are **open**—that is to say, when the eccentric radii point towards the link the rods are not *crossed*. In Fig. 171 the eccentric *does not lead* but follows the crank in the direction of rotation, because the steam port in the cylindrical slide valve fitted to the cylinder admits steam from between its inner edges, and not at the outer ends.

Separate Link.—In the arm of the weight shaft *G*, which is connected with the drag links, or bridle or suspension rods, as shown in Figs. 171 to 173, a slot is fitted with an adjusting screw and

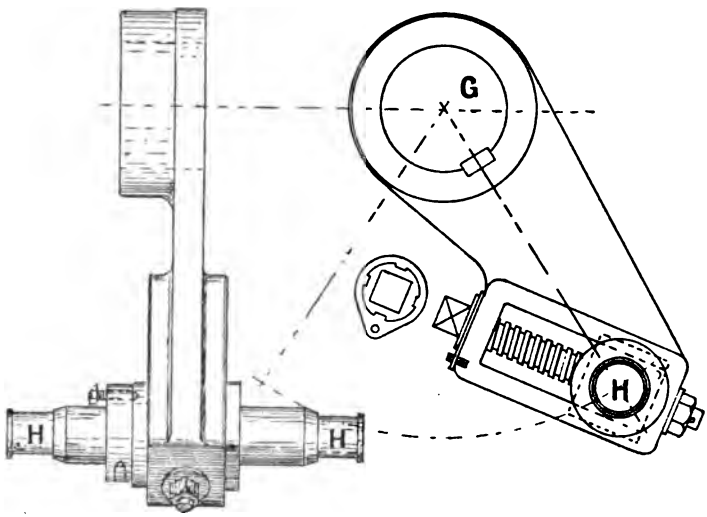


FIG. 173.—Independent Link.

block, *H*; the block forms the connection with the bridle rods through a pin joint. When the main link is in the proper position for ahead working, it is approximately in line with the slot in the arm, and a movement of the block, which is effected through the screw, draws the link over to any desired position. If the main link is in the astern position, the slot in the arm (or separate link)

is at right angles to the line of movement of the main link, and any movement of the separate link does not affect the relative position of the main link and its block, to which the slide rod is attached. Consequently the main engines can always be readily reversed, independently of the separate linking-up.

This means of subdivision is fitted in nearly all high-powered engines, and can be used at any time with safety and economy. Although an equal division of power may generally be considered as desirable, it does not follow that it gives the best economy—that can only be obtained with a due regard to the range of temperatures in the cylinders in conjunction with the actual coal consumption in practice.

At low powers, it is preferable to link up as far as possible consistently with smooth working, and at the same time to reduce the working pressure to the lowest safe limit for reversing and general purposes; this reduces the range of temperatures in the cylinders, and the loss consequent on cylinder condensation.

Independent Alteration of Cut-Off.—The weigh shaft operates each link in exactly the same way, and any linking-up so obtained tends to alter the cut-off equally in each cylinder. For lower powers this is not always satisfactory, and the effect of altering the cut-off in each cylinder independently will now be considered. In Fig. 174 the shaded portions of the figures represent the volume occupied by the steam in each cylinder in turn at the instant of cut-off. It may be considered that the same weight of steam occupies each cylinder in turn, for each piston makes the same number of strokes per minute.

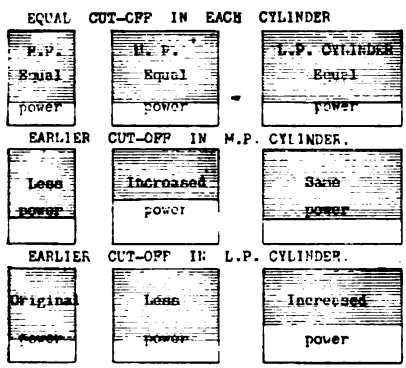


FIG. 174.—Effect of Alteration of Cut-Off.

1. *Cut-Off earlier in H.P. Cylinder.*—By cutting off earlier in the H.P. cylinder, a lesser volume of steam is admitted at each stroke, the expansion is greater, and consequently the mean pressure on the H.P. piston is less. The pressure admitted to the M.P. and L.P. cylinders is also less, and their mean pressures are less, and consequently there is a general reduction in the I.H.P.

and the number of revolutions of the engine. For lower powers, this method is generally used to obtain economy by using increased expansion.

2. *Cut-Off earlier in the M.P. Cylinder.*—The exhaust steam from the H.P. cylinder occupies less space in the M.P. cylinder, and consequently the back pressure in the H.P. cylinder is increased, and reduces the work done in that cylinder. The same weight of steam is admitted to the M.P. cylinder as before, but as it occupies less space, the pressure is greater. The exhaust from the M.P. cylinder is not affected, as it fills the same space as before, in the L.P. cylinder; and consequently, as the driving pressure is higher and the back pressure is the same, the mean pressure is greater, and more work is done in the M.P. cylinder.

3. *Cut-Off earlier in the L.P. Cylinder.*—On the same reasoning as for the M.P. cylinder, the work is decreased in the M.P. cylinder and increased in the L.P. cylinder.

It is thus seen that by linking up for any one cylinder or cylinders independently, a certain subdivision of power can be obtained, and for this purpose the separate, or independent, link is used.

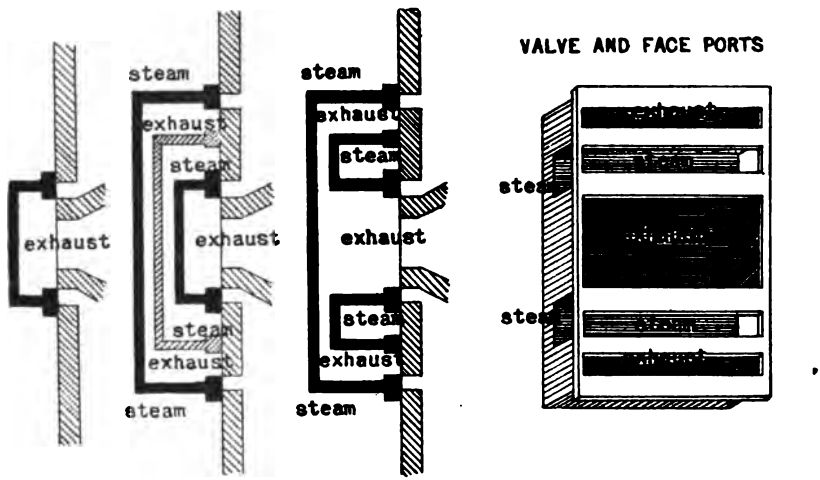
Double-ported Slide Valve.—The opening to steam through the cylinder port is governed by the movement of the slide valve, and the comparatively long stroke or travel of a single-ported slide valve requires a large radius of eccentricity and a correspondingly large eccentric, with a very large frictional loss. To reduce the travel and the frictional loss, the single-ported valve has been replaced by double or treble-ported valves, which give the same opening for about one-half or one-third the travel or stroke.

In the series of figures 175 to 178 the derivation of the double-ported valve from a single-ported valve is shown in stages. First, the single-ported valve is shown, with its two cylinder ports for steam and one for exhaust. In the next stage the original valve is partly covered by a diaphragm, which contains the steam pressure about the valve; outside this diaphragm a large valve is fitted, working over cylinder ports of similar width to the original ports fitted for the small valve. The three parts—the inner valve, the diaphragm, and the outer valve—are incorporated in one piece, and they travel together over the cylinder face, within a slide jacket or cover, which contains the steam pressure about the outer valve.

The necessary connection of the exhaust space outside the diaphragm, with the exhaust space within the inner valve and the cylinder exhaust port, is made by cutting away the back of the inner valve and part of the diaphragm, as shown in the third stage, and enclosing the steam ports of the valve.

The fourth stage shows the front face of a double-ported valve with its steam and exhaust ports and their connections, with the steam outside the valve and cylinder exhaust port, respectively.

All the operations of the double- or treble-ported valve are precisely similar to those performed by the single-ported variety.



FIGS. 175 to 178.—Principle of Multi-ported Slide Valves.

The double-ported valve (and treble-ported valve proportionately) takes up more room and increases the length of engine; there is greater wire-drawing, and consequent loss of velocity of the steam, from frictional resistance at the edges of the valve and cylinder ports, as their length is increased by the employment of multi-ported valves; but the reduction of travel with the decreased size of eccentric necessary, more than counterbalances these disadvantages.

In Fig. 179 a type of double-ported slide valve, as fitted to the L.P. cylinder of a large engine, is shown at about the middle of its stroke. The valve rod runs through the vertical middle of the valve, and is allowed a very slight play between the collar fitted on

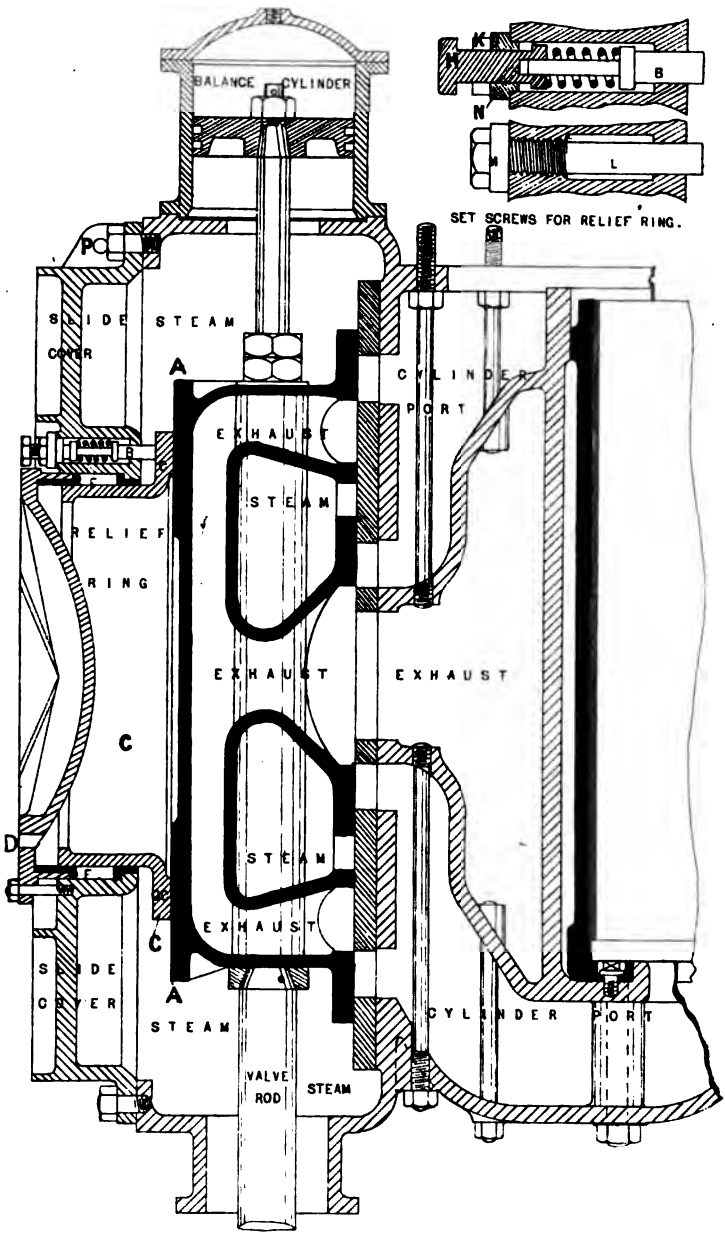


FIG. 179.—Double-ported Slide Valve.

a cone at the lower part of the rod and the check nuts on top of the valve; this amount should not be sufficient to allow "hammering," but it allows a movement of the valve towards the cylinder face, and the hole through the valve is therefore elongated in the required direction.

At the lower part of the valve chest a large opening is made for a gland and stuffing box (not shown) for the rod. Stays are fitted to some of the flat surfaces in the cylinder ports, and some of those at the top end also form the studs for securing the cylinder cover. The cover for the slide-valve chest or jacket is provided with lugs and holes, *P*, for supporting the weight of the cover when removed.

Relief or Equilibrium Ring.—A flat valve, whether single or multi-ported, is kept pressed up against the cylinder face by the steam pressure acting on the back of the valve, away from its face. When the pressure is great and the area of the valve is large, there is a great waste of work in overcoming the friction thus set up between the rubbing surfaces, and therefore means are provided to relieve the pressure from the back of the valve.

The back, *AA*, of the valve is made flat over as great an area as possible, with its surface parallel to that of the cylinder face; against this back face a packing piece, *C*, called the relief ring, and generally circular, attached to the slide jacket or cover, is pressed by small spiral springs and rods, *B*; the valve reciprocates between the cylinder face and the flange *CC* of the relief ring. The space enclosed within the relief ring is connected at *D* with the exhaust side of the valve through the receiver pipe, or, in the case of the L.P. valve, with the condenser. The pressure of the steam on the back of the valve is thus excluded from the area included by the relief ring, and reduces the total pressure on the cylinder face to a corresponding extent.

The simple method of connecting the space contained within the relief ring with the exhaust cavity of the valve is now found to be efficient in practice, but leakage is difficult to detect; the efficiency of the relief afforded can only be tested when the engine is at work. In the usual arrangement, a cock is fitted for detecting leakage by connecting it with a gauge and comparing the reading with the receiver or condenser gauge.

The relief ring is kept steam-tight at its junction with the slide cover by fitting a stuffing box and gland, *F*, packed with asbestos

or vulcanite rings, and in some cases Ramsbottom rings are used. In many of the older engines a bellows' ring was used; but with

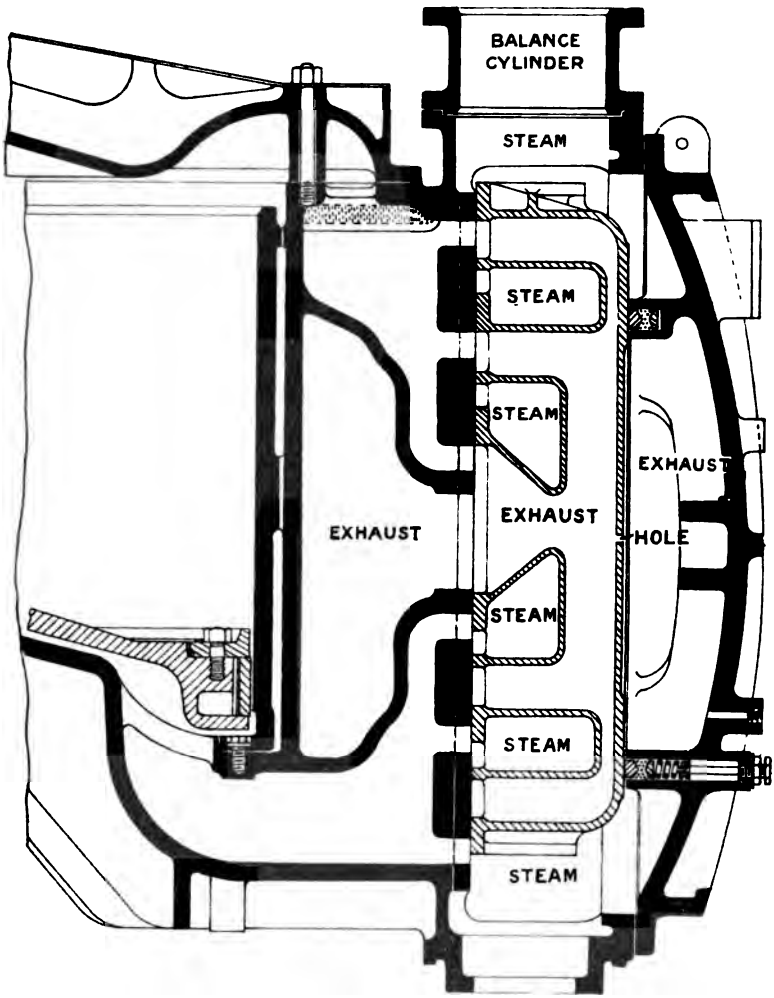


FIG. 180.—Triple-ported Slide Valve.

modern high pressures they were found to be very uncertain in their action under varying temperatures, and are now generally abandoned.

To prevent the valve being forced away from the cylinder face

by a momentary pressure, such as excessive cushioning of the piston, set screws (enlarged views of which are also shown in Fig. 179) and springs are fitted at the back of the packing ring to keep it pressed lightly against the back of the valve. The pressure exerted by the springs is very small, and the set screws, after being screwed up moderately tightly, are slacked back a certain amount, generally from $\frac{1}{32}$ to $\frac{1}{16}$ inch. In the event of the springs breaking, they can be set up more closely, but the slight play allowed generally prevents the springs from being broken. Two sorts of set screws are used: one with a spring, as in the upper figure; and one without a spring, as shown in the lower figure. They are generally fitted alternately round the relief ring. In the event of the spring breaking, the screw *H* can be pressed on the end of the spindle *B*, and secured in any required position by the lock-nut *K*. *N* is a gland-shaped piece, used to contain the screw *H*; after the spindle and spring are fitted in place, *N* makes a steam-tight joint with the valve-chest cover. The set screw *L* is accurately measured for the required length, leaving $\frac{1}{32}$ to $\frac{1}{16}$ inch play between its end and the flange of the relief ring, and a washer, *M*, is inserted to prevent any maladjustment. (See also Fig. 124.)

The gland at the back of the relief ring is generally made in the form of a cover for the space contained within the ring, as shown.

In Fig. 180 a somewhat different arrangement of the relief ring for a triple-ported slide valve is shown. Two wedge-shaped metal rings, with soft asbestos or other packing between them, maintain steam-tightness between the steam space and relief-ring space. A small hole connects the valve exhaust cavity with the relief space. The valve is shown in the position obtained just after the beginning of the up-stroke.

Cylindrical or Piston Slide Valve.—If an ordinary single-ported slide valve has its face curved round until it ultimately forms a cylinder, a cylindrical slide valve is produced of the type shown in Figs. 181 and 182. The length of the port is equal to the circumference of the valve, and approximately one-third the travel of the valve is required for the same opening to steam. In the flat valve the width of the port is assumed to be about equal to the diameter of the cylinder, and if the piston valve is of the same diameter, it is evident that the fore and aft length of engine must be increased. A diameter of valve about two-thirds the diameter of the cylinder is generally employed, and this also reduces the

clearance spaces, which are very large. With this diameter the same travel as the double-ported valve may be used.

The arrangement of double ports is usually too complicated for practical use; but two valves are sometimes used for the M.P. cylinder, and are then worked by the same link motion. This reduces the necessary length of engine, and has been adopted for several U.S.N. battleships, and in recent British warships.

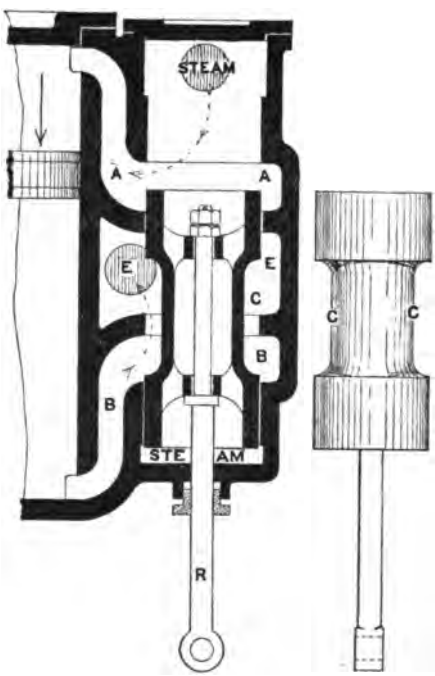


FIG. 181.—Cylindrical Slide Valve (end admission).

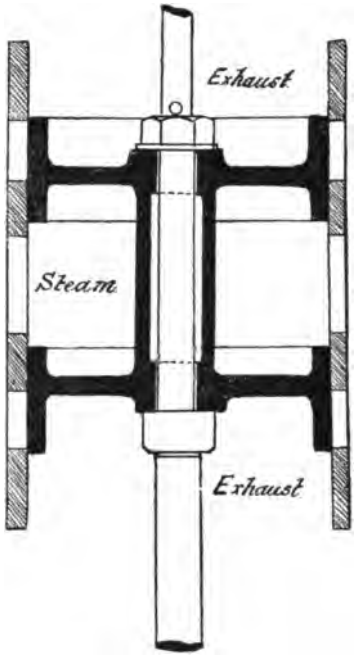


FIG. 182.—Cylindrical Slide Valve (central admission).

Cylindrical slide valves are generally fitted to the H.P. and M.P. cylinders, which are comparatively small, and any leakage which may take place is not of the same importance as leakage through the L.P. slide valve, because the steam is used in another cylinder. Many small engines are fitted with cylindrical slide valves only, and no loss from leakage is apparent, so that probably they will eventually supersede double-ported valves for L.P. cylinders.

The H.P. valve is frequently fitted without packing rings, as shown in Figs. 181 and 182, but in all other valves packing rings

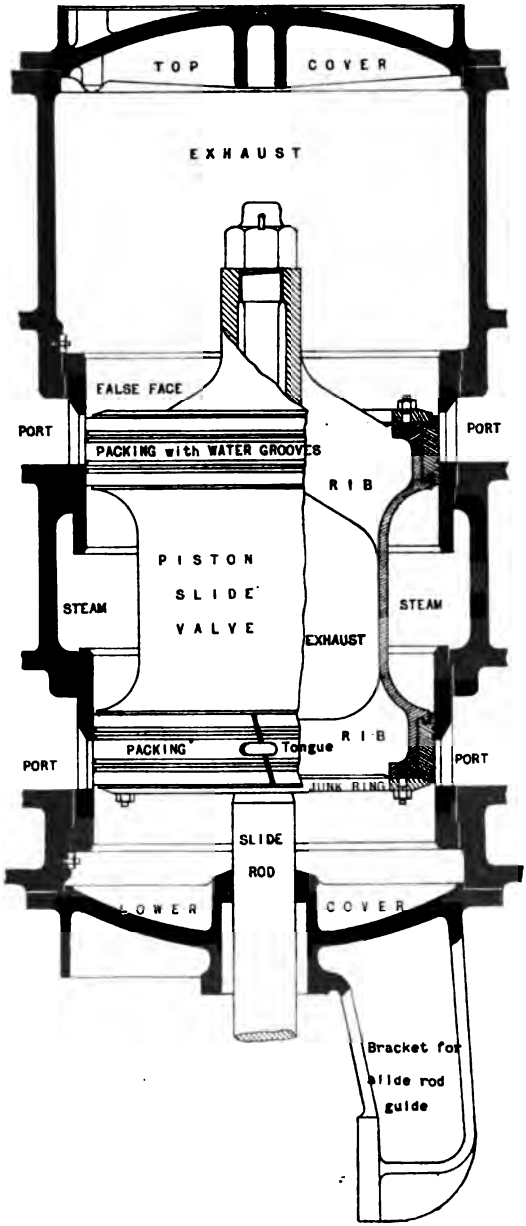


FIG. 183.—Cylindrical Slide Valve, with Packing Rings, etc.

are generally fitted. The steam is preferably admitted round the middle part of the valve, as shown in Fig. 182, because the valve-rod packing is then only subject to the lower exhaust pressure, which tends to decrease leakage and friction.

The valve moves on a cylindrical false face, which is secured, as shown in Fig. 183, in a similar manner to that in which the liner is fitted to the cylinder. The ports are generally ribbed diagonally, to prevent the packing rings being caught in them, but it is preferable to make the rings wider than the ports, as in Fig. 183, to prevent this occurring.

The packing rings are similar to those fitted to the piston, and a junk ring is also fitted, if required, for renewing or repairing the packing rings. Fig. 183 shows a cylindrical slide valve fitted with packing, junk ring, and tongue piece. The rings are restrained; this is usual for all packing rings fitted to slide valves.

No equilibrium or relief ring is necessary for a cylindrical slide valve, because there is no pressure on the false face except that exerted by the packing rings or springs.

If steam is admitted through the middle part of the valve, no balance piston is required to carry the weight, if the upper part of the valve can be made sufficiently greater in diameter than the lower part. When steam is admitted at the ends of the valve, a balance piston is necessary, and is generally fitted.

Balance Piston.—The weight of the slide valve and its connections, which are raised by the eccentric motion at each revolution of the engine, is very great, and the consequent friction between the eccentric straps and sheaves is objectionable. By fitting a small piston on the end of the valve rod above the valve to work in a small cylinder fitted above the valve chest (Fig. 179), the weight of the valve and its connections is supported by this piston. The upper side of the piston is connected with the exhaust from the slide valve to which it is attached, and its lower side is in connection with the steam on the back of the valve. The size of the balance piston is so proportioned that when the engine is working at full power the resultant total upward pressure on the piston is slightly in excess of the weight it has to support.

The balance piston acts as a guide for the slide-valve rod, and thus tends to keep the valve rod in proper position and working line; but as the hole through which the rod passes in the flat slide valve is elongated to allow the valve to be pressed against the cylinder face, this is not of much use to the slide valve except in extraordinary conditions. The lower part of the valve rod below the valve chest is generally fitted with a guide and block, so that there is only a direct lift on the link motion, as distinct from a bending action, which is otherwise produced.

Momentum Balance Piston.—This is a form of balancing arrangement, fitted in some cases where the valves are heavy, and therefore likely to bring a lot of work on the link motion when the slide valve reverses its direction of motion at each end of its stroke.

A sectional elevation of the momentum balance cylinder and piston is shown in Fig. 184. The piston is fitted on a continuation of the valve rod and immediately above the valve. The upper part of the cylinder *A* can be connected, if required, with the exhaust steam through the pipe *C*. The lower part of the cylinder

below the piston is connected with the steam side of the valve through an annular space about the rod, provided by making the neck bush larger than the rod.

On the down-stroke of the valve, the steam is imprisoned below the piston by the closing of the connection at the neck ring, where a spigot or projection on the piston descends into the space about the rod. The imprisoned steam forms a cushion, and takes the momentum of the lowering weights and brings them gently to rest, and at the same time accumulates sufficient pressure to assist the reversal of direction of motion.

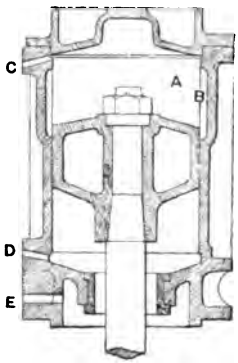


FIG. 184.—Momentum Balance Piston.

On the up-stroke, the steam is also imprisoned, and a cushion is formed, but the accumulated pressure is moderated by allowing some of it to escape through two grooves, *B*, made in the sides of the cylinder near its upper part, just previous to the top of the stroke. If the pressure above the piston is less than the steam pressure, there is access through the grooves, and sufficient enters for the necessary cushioning, which commences after the openings are closed by the upward motion of the piston.

In the event of too great cushioning and resulting shock, the space above the piston can be connected with the exhaust by opening a cock at *C*; and the space below the piston can be connected by opening a cock fitted in a pipe connecting up *D* and *E*.

Joy's Assistant Cylinder.—In this arrangement (Fig. 185) the balance piston tends to relieve the link motion of all work except the initial guiding motion. For large engines and valves the assistant cylinder may develop as much as 25 I.H.P., and by the application of the indicator an exact reading of the action inside the cylinder can be made and any calculations checked. The balance piston is sometimes kept moderately steam-tight in its cylinder by two or more Ramsbottom rings, each of greater width than the steam ports which they travel across.

Steam is admitted to an annular space in the cylinder at about its mid-length; the piston overlaps this space at all parts of its stroke. Steam ports are fitted in the piston to admit steam as required either above or below it, and the opening is regulated by

the movement of the piston. The connection with the exhaust is also opened by the movement of the piston, and exhaust ports are made in the cylinder, which are uncovered at the correct instant of the stroke. The steam and exhaust ports are arranged so that there is no passage-way between them when the piston is in place.

For the up-stroke, steam admission begins at about one-tenth before the end of the down-stroke, and is cut off when the piston reaches one-tenth of its up-stroke, and is expanded until the piston uncovers the exhaust at about six-tenths of the stroke, when release begins. Above the piston the exhaust is closed at about

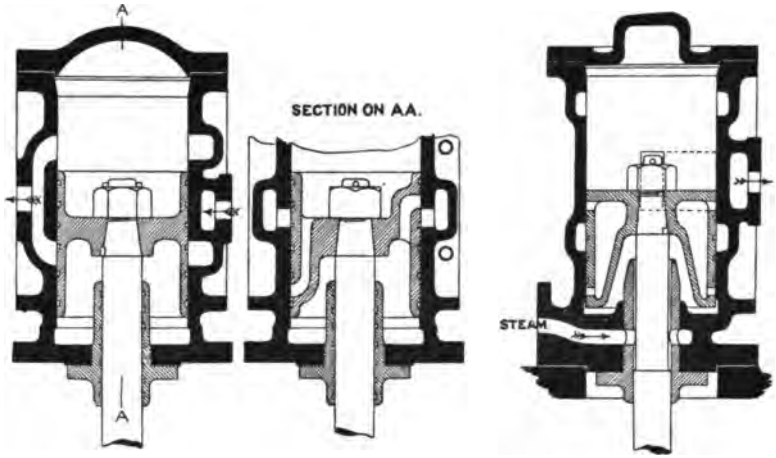


FIG. 185.—Joy's Assistant Cylinder.

four-tenths of the stroke, and compression then begins, and continues until admission begins for the down-stroke, at about nine-tenths of the up-stroke. For the down-stroke the actions are similar; admission (or lead), cut-off, expansion, release, and compression are all repeated.

For naval engines, which are frequently worked with the link considerably shortened, this arrangement may become inoperative if the travel of the piston is insufficient to admit steam to the assistant cylinder at each stroke.

Brown's Reversing Gear.—This arrangement for reversing the engine is fitted in a great number of ships of the mercantile marine, and the type shown is very similar to that fitted in the *Kaiser Wilhelm II.* It is also used for operating the manœuvring valves of the turbine engines of the *Mauretania*, etc.

REFERENCES TO FIG. 186.

A, Steam cylinder, in which piston works connected with rod *B*.
B, Piston rod, common to both pistons, steam and hydraulic.
C, Hydraulic cylinder, in which a piston works attached to the common piston rod *B*, packed with double packing leathers, which open to pressure above or below the piston, and maintain pressure tightness.
D, Crosshead on piston rod.
EE, Rods connecting crosshead with reversing arm of weigh shaft.
F, Control valve for hydraulic cylinder, worked by rod common to slide valve of steam cylinder.
G, Tank containing mixture of water and glycerine, connected with control valve and with bottom and top of hydraulic cylinder through a three-way cock.
H, Reversing lever, with spring attachment for setting lever in sector.
K, Sector.
L, Fulcrum of lever *DLM*.
M, Rod connecting the lever *DLM* with the common valve spindle *N*.
N, Valve spindle, connected with both hydraulic control valve and steam slide valve.
O, Curved link rigidly attached to lever *DLM*.
P, Block carried by crosshead *D* and sliding over link *OO*.

The reversing gear can be moved by steam, by pulling the lever *H* into the "ahead" position, or by pushing it into the "astern" position. When required to go ahead, the fulcrum *L* is raised by moving the lever *H*; the connecting rod *MN* is thus moved upwards, and the slide valve with it; steam is admitted below the piston, and raises the reversing arm towards the ahead position. As the pistons rise, the block *P*, attached to the crosshead *D*, tilts the lever *DLM* about the fulcrum *L* (the vertical movement of which is restricted by the reversing lever *H*), and returns the slide valve to its central position, thus bringing the steam piston to rest.

The movement of the buffer piston in its cylinder is controlled by the valve spindle *N*, which is also attached to the water-control valve. When the steam valve is closed, the control valve is also

closed, and as no water can then pass from one side of the buffer piston to the other, any movement of the steam and water pistons is checked.

The reversing arm of the weigh shaft can be stopped in any

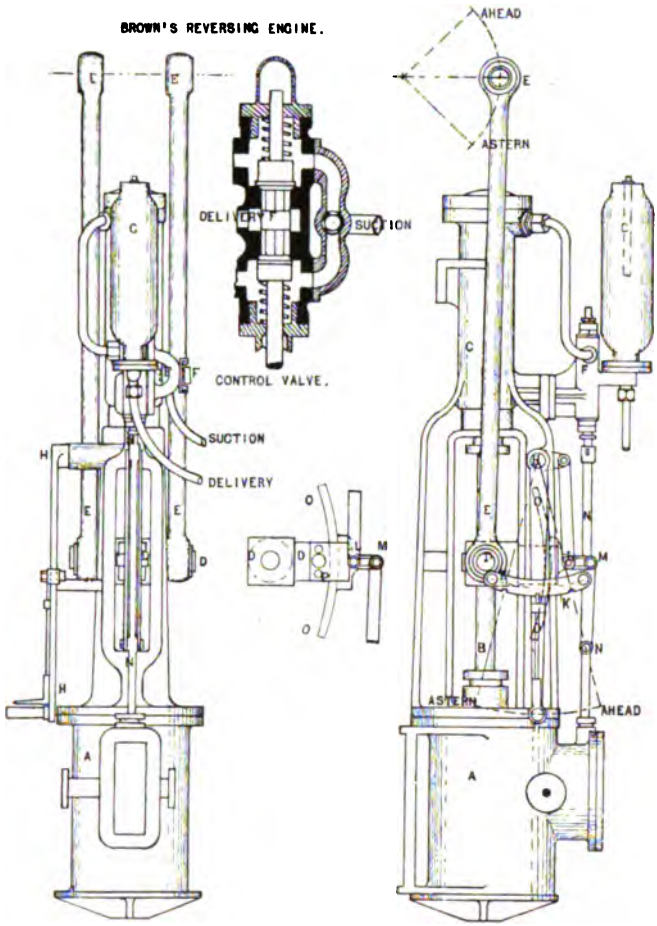


FIG. 186.—Brown's Reversing Gear.

intermediate position, for linking up, by returning the reversing lever *H* into the middle position as required by hand, and thus closing the cylinder port to steam and closing the control valve.

When steam is not used and hand-gear is required, the

reversing arm of the weigh shaft is moved into any desired position by using a hand-pump (hydraulic) and manipulating the three-way cock. Water is drawn into the pump from the tank, which is connected through the three-way cock with the proper end of the cylinder; and by setting the reversing lever, as for steam working, pressure is admitted through the control valve to the opposite end of the cylinder, and moves the piston into the required position.

The control valve is the solid piston *F* on the rod between the two valves shown in the enlarged figure. The rod passes through the centre of the valves and does not move them; they are returned to their seatings by spiral springs.

Joy's Valve Gear.—In this arrangement the movement of the slide valve is derived from the connecting rod without the use of eccentrics. In Fig. 187 the connecting rod is represented by *ABC*, and the lever *DEF* operates the valve through the fulcrum *E*. The end *F* of the lever is attached by a connecting rod to the slide-valve rod, and the other end, *D*, is attached to the connecting rod *ABC* by the links *BG* and *GH*; where *B* is a pivot on the connecting rod *ABC*, and moves with it, and *H* is a fixed pivot on the engine framing, and is the only fixed axis in the gear. The fulcrum *E* of the lever *DEF* is constrained to move along *PP'*, or *QQ'*, as required for ahead or astern working. In the figure, *E* is suspended to a rod, the point of suspension of which rod can be altered, as shown in dotted lines. (In Hackworth's gear the curve *PP'* is a slot, along which the point *E* is constrained to move; and the inclination of the slot can be altered to *QQ'* for astern working. In Marshall's gear a suspension link is used, as shown in the figure, and the point of suspension is altered to obtain astern working.) Intermediate positions correspond to various degrees of linking up.

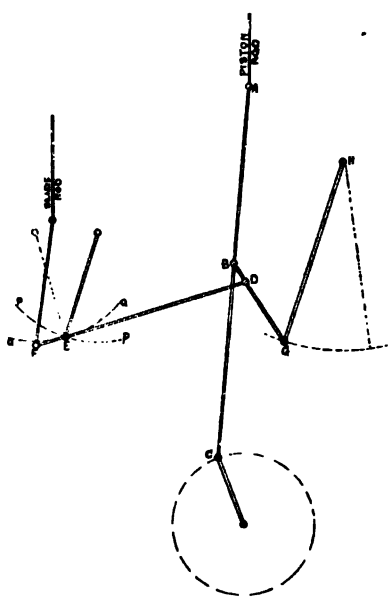


FIG. 187.—Joy's Valve Gear.

Two independent motions are given to the point F:—

1. The reciprocations of the piston are transmitted to F by the action of the lever DEF. This part of the motion is constant, and independent of the position of the gear, and the proportions are chosen so that its amount is equal to the *lap* added to the *lead*. In other words, the ratio of EF to ED is made equal to the lap *plus* lead, divided by half the stroke of the piston.

2. The movements of the connecting rod ABC are transmitted to the fulcrum E, and cause it to move along the curve PP', or QQ', according to the position of the gear for ahead or astern working. This additional portion of the valve's motion, obtained from the movement of the connecting rod, provides for the port opening beyond the amount given by the *lead*, and can be regulated by altering the angle of inclination of the curve PP'. This alteration provides for astern working and for various degrees of cut-off and expansion; all the operations of the slide valve are satisfactorily obtained by this valve motion.

If one end of the lever FED were attached directly to the connecting rod at B, the varying obliquity of this lever itself would cause the fulcrum E to move slightly to and fro on the curve; to avoid this, and to secure that the movement of the fulcrum on the curve should be derived from the movement of the connecting rod ABC only, the parallel motion $EDBGH$ is fitted. This motion is so proportioned that if the point B be moved along BC in a straight line, the point E remains stationary; therefore the disturbing effect of obliquity of FED is avoided.

There are many other valve gears where the movement of the valve is derived from that of the connecting rod, or from one eccentric, such as in Bremme's. The objection to them generally is the number of pin joints and working surfaces, which are likely to produce lack of uniformity in the motion of the slide valve after they are somewhat worn. The advantage apparent from fitting the slide valve in front of the cylinder, as shown in Fig. 187, is a shorter engine, with a slight saving in weight; it is more than probable that this advantage outweighs any of the objections that can be raised.

Geometrical Representation of the Motion of the Slide Valve.—Take XX' and YY' at right angles to each other, as in Fig. 188, and intersecting at A; from the centre, at radius equal to the half-travel of the slide valve, describe a circle, XCD. From any point

B in the line XX', at radius BC, equal to the length of the eccentric rod, draw an arc of a circle, CED, cutting the circle XCD at C and D, and cutting AB at E. Then the point E moves in its path XX', which represents the travel of the valve, exactly as the valve, as represented by the point B, moves. (The proportion of AC to CB is much exaggerated in the figure to show the effect of obliquity more clearly.)

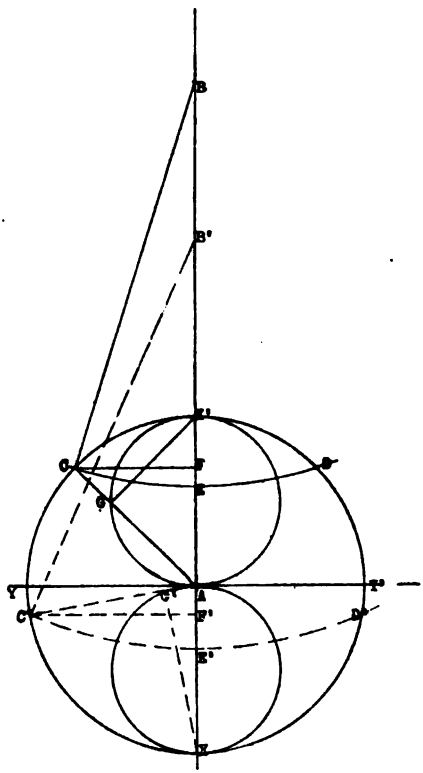


FIG. 188.—Geometrical Motion of the Valve.

The eccentric rod is generally very long compared with the eccentric radius—eight or more times; and if it be indefinitely long, the position of the valve in its path may be represented by a point, F, where CF is drawn perpendicular to AX. The length EF is very small, and may be neglected; technically, this is termed neglecting the obliquity of the eccentric rod.

Set off AG equal to AF, that is, the distance of the valve from the middle of its stroke. Join GX'. In the triangles CAF and $X'AG$, AG is equal to AF, and AC to AX', and the angle CAX' is common to both; therefore the angle AGX' is equal to the angle AFC, that is, to a right angle.

If various positions of F be laid off, it is seen that G lies on a pair of circles, having AX' and AX as diameters. Thus, the radial distance AG of any point, G, on these circles from A, represents the distance the valve is from the middle of its stroke, for the same position of the eccentric radius AC.

Next, suppose the engine crank is on one of the dead centres, AX (Fig. 189), and rotating in the direction of the arrow. Then the eccentric radius is in a position, AC, such that YAC equals the angle of advance, and the position of the valve is represented by M,

the foot of the perpendicular, CM to AX. Take AG, equal to AM. Now, suppose the crank to turn into a new position, AC', through any angle, a, the eccentric radius will move through an equal angle, CAC'', and the valve moves from M to M''. Along AC' take AG'', equal to AM'', and continue the construction for any other positions.

Then G is found to lie on a pair of circles, each of whose diameters is equal to the half-travel of the valve, *i.e.* equal to AX. One of these two circles touches the circle XPC, whose radius is equal to the half-travel of the valve at P. Now, in the triangles $AG'P$ and AMC, AC is equal to AP, and AG' was made equal to AM, and each of the angles AMC and $AG'P$ is a right angle; therefore the angle PAX is equal to the angle CAX'. The angle PAY and angle CAY are also equal, because they are the complements of the angles PAX and CAX'. That is, the angle PAY is equal to YAC, the *angle of advance*.

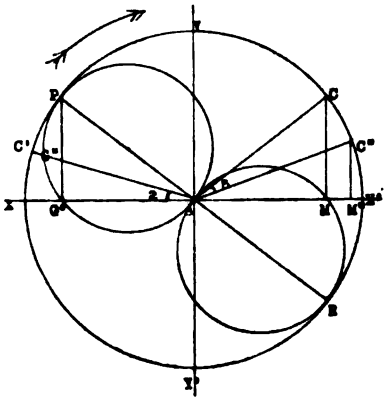


FIG. 189.—Geometrical Motion of the Valve.

Obliquity of the Eccentric Rod.—Reference to the above figures (188 and 189) shows that obliquity cannot be neglected when setting the slide rod and valve practically. In the case of a vertical inverted direct-acting engine, the tendency is to increase the mean downward pressure and to diminish the mean upward pressure; this, added to the weight, tends to produce an irregular motion of the engine.

To overcome this tendency, the slide rod is lengthened so that the centre of the valve in the middle of its stroke is a little higher than the centre of the exhaust port in the cylinder face. This increases the steam lap at the top and diminishes it at the bottom, but the lead diminishes at the top and increases at the bottom. The sum of the two laps, the sum of the two leads, and the mean cut-off, remain as before. Thus for the general purposes of designing slide valves for various engines, and the graphical measurement of the various functions, the method, neglecting obliquity, retains all its usefulness.

Zeuner's Diagram.—All the *operations* of the valve may be represented geometrically by means of the Zeuner valve circles, shown in Fig. 190.

Take XX' and YY' at right angles to each other and intersecting at A. Through A draw PAR so that the angle YAP is equal to the angle of advance. Make $AP = AR =$ radius of eccentricity, so that PR is equal to the travel of the valve. Describe circles on AP and AR as diameters. Describe a circle with AP as radius to represent the path of the crank pin. Describe circles NNN, with a radius equal to the steam lap, and LLL, with a radius equal to the exhaust lap, from the centre A.

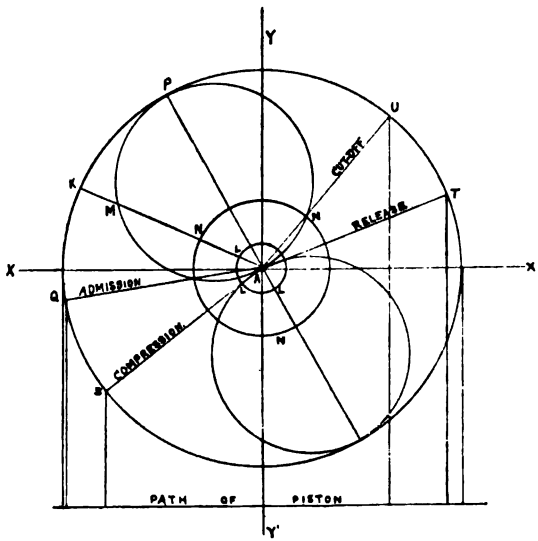


FIG. 190.—Zeuner Circles.

For any position of the crank AK, where AK cuts the valve circle AMP at M, the distance the valve has travelled is represented by AM. MN is the opening of the port to steam, and ML is the opening of the exhaust port.

Similarly, AQ is the position of the crank when admission begins;
AU „ „ cut-off takes place;
AT „ „ release begins;
and AS „ „ compression begins.

If there is no exhaust lap, AT and AS will be at right angles to PAR.

When linked up, a very approximate diagram of the motion and operations of the valve can be obtained by the application of the above constructions.

First construct the diagram for working in full gear (Fig. 191), where AP represents the half-travel of the valve. When the link

is middled the travel of the valve is approximately represented by *twice the length of the perpendicular let fall from the centre of the shaft on the line joining the centres of the eccentrics*; set off this length along *AX*. Through the point found above, and through *P*, draw an arc of a circle, *PBP*, with a suitable radius and centred on *AX*, as shown. *PBP* represents the length of the link, and by setting off *PB* so that it represents to scale the position of the block when linked-up, *AB* is the maximum travel of the valve from its central position.

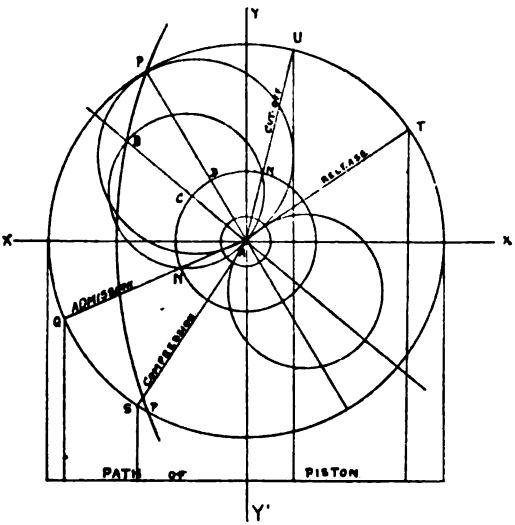


FIG. 191.—Zeuner Circles.

In Fig. 192, suppose the block is in some position, *Z*, between *L* and *K*; then—

$$PB : PP \text{ (Fig. 191)} = LZ : LM \text{ (Fig. 192)}.$$

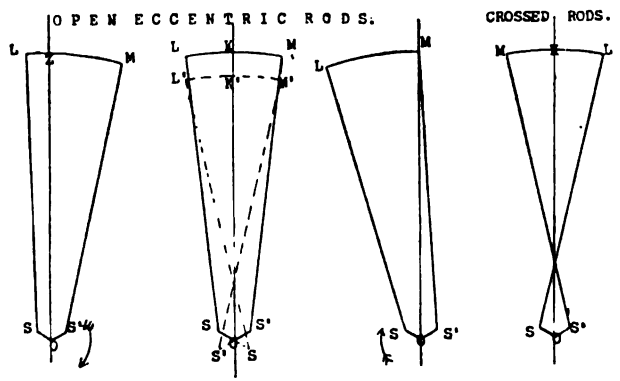


FIG. 192.—Open and Crossed Rods.

The construction for the valve circles is shown in Fig. 191, and also the positions for the various operations of the valve when linked-up.

In cases where the link is *overhung, as with the solid-bar link,* the points L and M (Fig. 192) are always outside the extreme positions of the link block, and consequently the gear is always linked-up. The construction of the valve circles is obtained by first constructing those theoretically due to the full travel of the eccentric radii, and then following the procedure shown above for the valve when the link is shortened.

Or, when the travel of the valve is known for both full gear and in mid-link, the curve can be found as above; call this PP in Fig. 191, then take a point, D, outside PP, such that—

$$\frac{PD}{PP + 2 \,.\, PD} = \frac{\text{distance between end of link and full gear}}{\text{full length of link between centres}}.$$

PART V

THE CONDENSER, STEAM, AND FEED WATER SYSTEMS

CHAPTER XVII

CONDENSER, AIR PUMP, AND CIRCULATING PUMP

Introductory.—One pint of water makes a *relative volume* of 1643 pints of steam under ordinary atmospheric pressure, or 11,000 pints under a pressure of 2 lb. absolute, or 20,700 pints under a pressure of 1 lb. absolute, which is equivalent to about 28 inches of vacuum by the gauge.

Therefore if 20,000 pints of steam be suddenly and entirely condensed in a condenser where a vacuum of about 28 inches is obtainable, they would fill only about one pint as water; and the remaining space or volume contains practically no vapour, with a consequent very low pressure or tension in the space of 19,999 pints. Steam is a readily expanding fluid, and the exhaust steam from the engine is not all and entirely condensed in the cooling chamber; a very small quantity expands as the pressure falls, and, expanding as vapour, fills the space left by the condensation of the remainder. A small proportion of air and uncondensable vapour also enters with the steam; this prevents the formation of a perfect vacuum, as well as some purely mechanical causes.

The reduction of the exhaust pressure by using a condenser decreases the back, and therefore retarding, pressure on the *LP* piston, and thus increases the mean pressure and power obtainable from the engine, without requiring an increased quantity of steam from the boiler. Economy of steam for an equal power is thus

obtained by using a condenser, and a further economy is also obtainable from the possibility of using a greater ratio of expansion with a decreased back pressure.

Jet Condenser.—With a jet condenser the steam is condensed by mixing the cooling water, in the form of spray, with the steam. For each 1 lb. by weight of steam condensed about 40 to 60 lb. of cooling water are required; and if the cooling water is taken from the sea, the mixture produced is thus practically sea-water. This mixture when used as feed water leaves a lime scale on the heating surfaces of the boilers, and reduces their efficiency. An accumulation of scale is dangerous in high-pressure boilers, and therefore for marine purposes the jet condenser has been generally and almost entirely abandoned.

Surface Condenser.—For sea-going steamers, and for land engines wherever possible, surface condensation in which the cooling water and condensed steam remain separated, is almost exclusively used. In addition to the advantage of the jet condenser in reducing the back pressure on the L.P. piston (with a contingent increase of expansion), all the condensed water—except a small percentage of loss from leakage—is saved and returned to the boiler as feed water, uncontaminated by sea-water.

In Fig. 193 *H* is the condenser body, at each end of which is a flat tube plate jointed to the body by flanges formed on the outer casing. The bolts and nuts used for these joints also serve for the joints of the end covers. The tube plates are generally about 1 inch thick, thinned down to about $\frac{3}{4}$-inch at the joining surfaces. Holes are drilled through them, and a great number of small pipes, or tubes, are fitted from tube plate to tube plate and made tight at their ends in the holes by packed glands. The internal passages through the tubes connect the two end spaces, between the tube plates and the end covers, with each other. The surfaces of these tubes constitute the cooling surface of the condenser, and a certain surface area is provided in the condenser according to the weight of steam which is required to be condensed per hour for the total horse-power.

The cooling water enters from the sea by the main inlet *D* and sea suction valves, passes through a pipe, *E*, into the circulating pump *F*, and is driven by the pump through the pipe *G* into the space at the end of the condenser between the tube plate and cover. Thence it passes internally through the tubes to the other end of

the condenser; and, in the arrangement shown, back through the upper rows of tubes to the front end, and overboard through the pipe *V* and the outlet valve. The direction of circulation is shown by the arrows. The water space at the front end is divided horizontally, so that the water cannot take a short cut overboard;

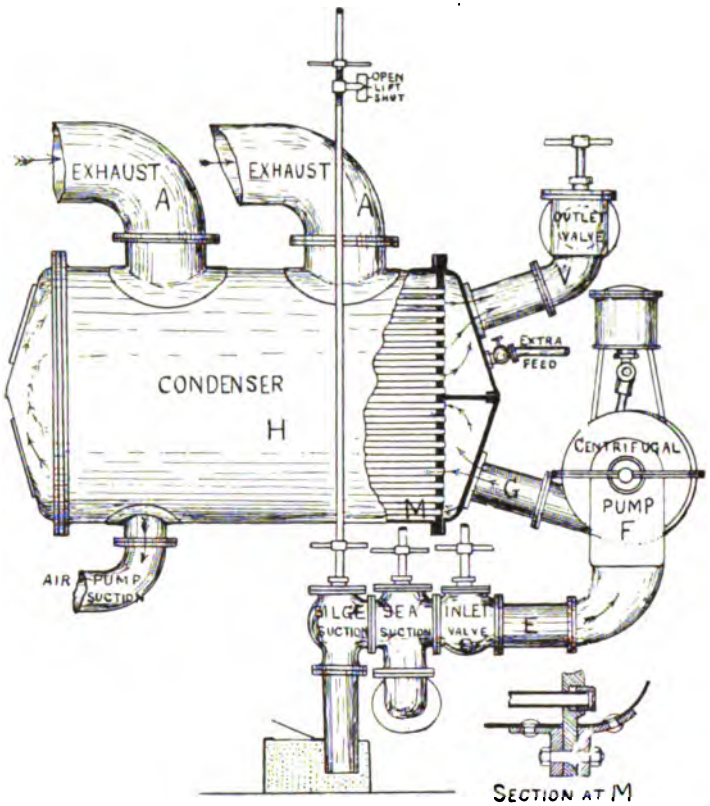


FIG. 193.—Surface Condenser.

and in this plate a valve is now fitted, as shown in Fig. 195, which can be opened when the circulating pump is used to pump out the bilge—so that the dirty water need not pass through the tubes,—and to increase the efficiency of the pump. (The division plate is sometimes fitted vertically, when the inlet and outlet are fitted near each other, as shown in Fig. 195.)

The *covers* should be shaped at the inlet and outlet openings to deflect the water equally over the tube plate, so that each tube

should do its share of the cooling; the covers are now generally adapted for this purpose (Fig. 195). Small handholes, about 10 to 12 inches in diameter, are fitted in the covers for facilitating examination of, and small repairs to, the tube ends. The covers are usually made with webs, to give them sufficient stiffness and strength. The tube-plate joint is shown in Fig 193A, and it should be noticed that the cover can be removed without disturbing the joint between the body of the condenser and the tube plate.

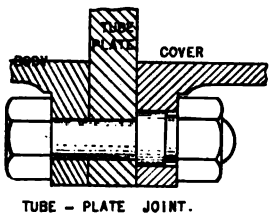


FIG. 193A.—Tube-plate Joint.

In a few cases the steam passes through the tubes internally, while the cooling water is circulated externally to the tubes. This arrangement has one advantage, in that leakage at the tube ends can be more readily detected, and the test pressure is applied in the same direction as when the condenser is at work. There are some disadvantages, however, generally of a practical nature, and in the majority of cases the condensers are arranged as fully described and illustrated in this chapter.

A direct tube arrangement is sometimes fitted, particularly in small vessels, when the water passes through the tubes in one direction only and then directly overboard; a division plate is then unnecessary.

Path of the Exhaust Steam.—Referring to Fig. 195, which shows a modern type, the steam enters the condenser from the eduction pipe at the top of the casing and strikes a baffle; and, dribbling down externally to the tubes, becomes condensed on the cooled surface of the tubes during its passage to the bottom, whence it is removed by the air pump. The connections and mountings fitted to the main condenser on the steam side are:—

At the top—

1. Vacuum, and combined pressure and vacuum gauges.

2. Eduction pipe from main engine. In recent naval vessels there are two main condensers and two L.P. cylinders to each main engine, and a valve is fitted, as shown in Fig. 207, close to each condenser in the eduction pipe common to both condensers and L.P. cylinders.

3. Exhaust from reversing engine and one main circulating engine.

4. Silent blow-off. In some recent arrangements the automatic blow-out or separator drain is used as a silent blow-off.

(Numbers 3 and 4 are connected with the eduction pipe in the arrangement shown in Fig. 207.)

At the bottom—

1. Main air-pump suction.

2. Auxiliary air-pump suction, or hot-well pump suction, if fitted. If independent air pumps are fitted, neither of these arrangements is required.

3. Connection with the reserve tanks, to draw water from them to *make up* the feed-water loss.

4. Drain from L.P. cylinder fitted with a baffle, as shown in Fig. 194. If this drain is led into the crank pit, bilge, or feed tank, water may be drawn up *into* the cylinder and damage done.

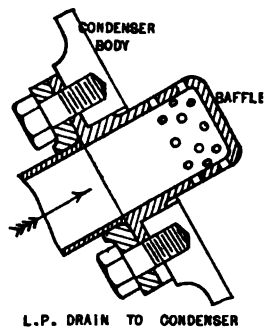


FIG. 194.—Inlet Baffle.

The connections and mountings fitted to the auxiliary condenser are :—

At the top—

1. Combined pressure and vacuum gauge; sometimes two are fitted.

2. Auxiliary exhaust pipe.

3. Exhaust from the auxiliary air pump and one main circulating engine.

4. Pipe conveying the *gained steam* from the evaporator, when used for boiler *make-up*.

5. Exhaust from steering and dynamo engines. Separate pipes are generally fitted for these engines to overcome the tendency to fluctuation of speed from a varying exhaust pressure.

At the bottom—

1. Air-pump suction.

2. Drains from the H.P. and M.P. cylinders and slide-valve chests fitted with baffles, as shown in Fig. 194; by means of switch cocks these drains can be led into the bilge.

3. Drains from the main engine cylinder jackets. In some cases these drains are led into the feed tank or hot well, and some economy is effected by heating the feed water.

4. Drain from the *primary* steam of the evaporator coils. In some cases the connection is made with the top of the condenser.

(When no auxiliary condenser is fitted, the connections are generally made with the main condenser, through the eduction pipe, shown in Fig. 207.)

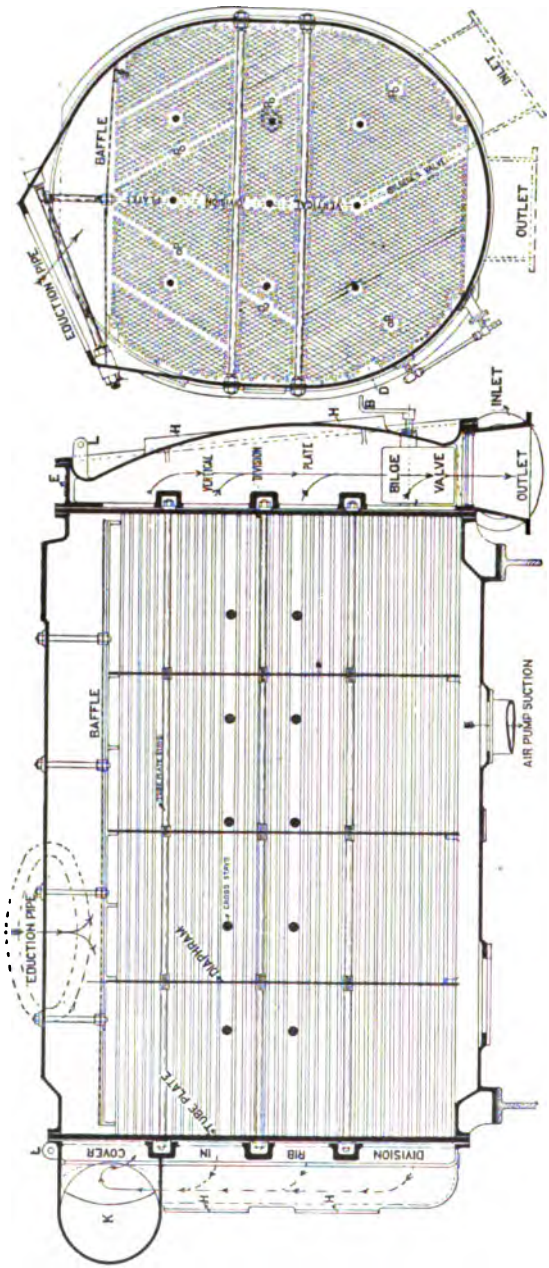


Fig. 195.—Modern Type of Surface Condenser.

Baffles.—All inlets to the steam side of the condenser are covered by baffle plates, secured to the casing, to absorb the momentum of the entering steam, and thus prevent any violent or direct impingement on the tubes, which may produce vibration, and consequent abrasion or splitting. The baffle plate for the main eduction pipe is shown in Fig. 195, and is strongly held to the casing, and perforated.

Condenser Tubes.—For naval purposes the tubes are made of gun metal of special composition, and tested to an internal pressure of 300 lb. per square inch. The composition generally adopted consists of at least 70 per cent copper, 1 per cent tin, and the remainder zinc. The casing and all parts of the condensers are made of gun

metal of as nearly the same composition as the tubes as possible, with not less than 1 per cent of tin.

For naval purposes the tubes are $\frac{5}{8}$-inch diameter externally, $\frac{1}{20}$-inch thick, and the length is now standardised to some multiple of 5 feet. They are pitched at the corners of an equilateral triangle, at a distance of $\frac{31}{32}$-inch to 1 inch from centre to centre.

For the mercantile marine the tubes are sometimes made of pure copper, and are generally $\frac{3}{4}$-inch in diameter.

The surface area of a tube $\frac{5}{8}$-inch diameter and 5 feet long is about 0·818 square foot; and about 1 to 1·2 square feet of cooling surface being allowed per I.H.P., the number of tubes and glands is very great.

Tube Glands.—The tubes are passed through holes in the tube plates, and when in place the ends are about level with the outer faces of the plates. A screwed gland is inserted into the plate over the end of each tube, as shown in Fig. 196, and a cotton

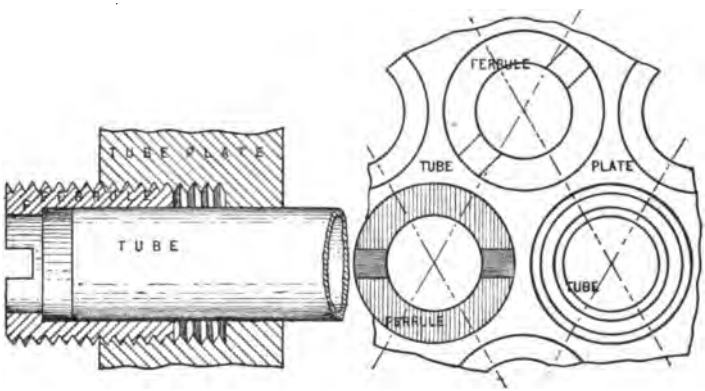


FIG. 196.—Tube Packing Gland.

grommet under the gland makes a tight joint with the tube plate. The glands are made with an inner flange, so that the tube cannot work through them, and about $\frac{1}{16}$-inch clearance is allowed for unequal expansion between the end of the tube and the flange. A slot is cut across the end of each gland for use when screwing up or removing. The glands are about $\frac{3}{16}$-inch diameter externally for a $\frac{5}{8}$-inch tube.

Condenser Stays.—The tubes do not support the tube plates, they merely form the connection between the outer sides of the

plates for the water to pass through the tubes forming the cooling surface without mixing with the condensed steam. The outer, or circulating water, side of each plate is subject to a pressure, or head of water, due to the depth at which the condenser is placed below the water-level outside the ship; the inner side of the plate is subject to less than atmospheric pressure in proportion to the vacuum, and thus the tendency to collapse is augmented on the plates, and a total pressure of about 30 lb. per square inch must

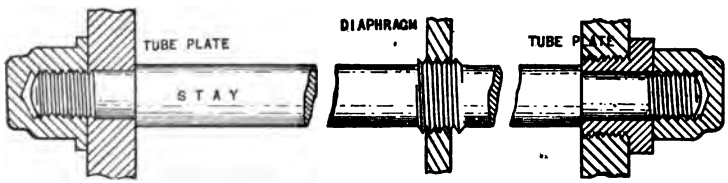


FIG. 197.—Condenser Stays.

be provided for. Stays are therefore fitted from plate to plate, at proper intervals, and in the place of some tubes which are consequently omitted. A form of stay, which can be removed without disturbing the tubes, is shown in Fig. 197; the hole in the front plate is large enough to allow the withdrawal of the shoulder fitted to support the plate at each end. In many cases the stays were simply fitted with check nuts on opposite sides of the plate, but these could not usually be removed or set up without removing the tubes in the vicinity.

If the condenser is not circular in shape, and some ground space is saved by making it flat-sided, the flat sides must be prevented from bulging. Stays are therefore fitted across the condenser body, as shown in Fig. 195; and they are conveniently made of the form described above, with bosses on the casing, to give the necessary local strength for the nut and screw thread.

Diaphragm Plate.—When the condenser tubes are over 5 feet in length, it is necessary to support them, so that the vibration and general sagging does not bring them in contact with each other. Supporting plates, or diaphragms, are therefore fitted not more than 30 inches from each other, or from the tube plates, through which the whole of the tubes are reived. The necessary holes in the diaphragm plate should be a moderately good fit for the tubes passing through them, or the vibration within the holes causes abrasion. In some cases india-rubber rings are fitted over the tubes,

to absorb the vibration and to keep the tubes properly apart. If the diaphragm plates are fitted exactly at equal distances the vibrations tend to synchronise, and increase in intensity; by fitting the plates at unequal distances the vibrations tend to decrease in intensity. In Fig. 195 the tubes are about 10 feet long, and the distances between the diaphragm and tube plates are $28\frac{3}{4}$, $28\frac{1}{2}$, 28, and 30 inches respectively.

A **weed trap** is fitted to nearly all inlet pipes for the circulating water where the inlet is not very deeply immersed, as in destroyers and for auxiliary condensers. It is generally fitted at *E* (Fig. 193), so that, by removing a cover, any weed that may have been caught can be removed, and in most cases the grating can be taken out and cleaned. The inlet and outlet valves should be closed when clearing the weed trap, and of course the steam must not be exhausted into the condenser while the water is shut off. Sometimes the direction of flow of the circulating water can be reversed, and the weed thus cleared from the grating and tubes; or a fire-hose can be connected with the pump to effect the same purpose.

Protecting plates are fitted in the water spaces at the ends of the condenser, in electrical connection with the tube plates. The plates were made of zinc, then cast iron was tried, and they are now almost invariably made of mild steel. Protection plates are also fitted in the bottom part of the condenser on the steam side and to the weed-trap grating, when one is fitted. If the plates are secured to the handhole covers—until recently a common practice — care must be taken that they are not insulated from the tube plates by the jointing material. A modern method of fitting steel protectors is shown in Fig. 198. It secures electrical contact with the tube plates at all times.

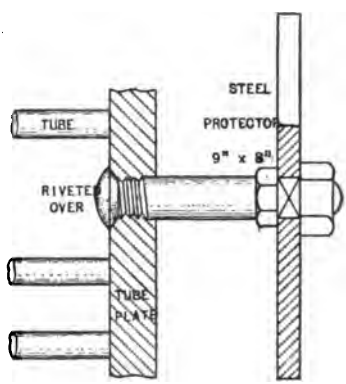


FIG. 198.—Steel Protecting Plate.

The *failure of condensers* is generally due to leakage from either mechanical or corrosive causes. The mechanical causes are generally (1) inefficient baffling, with a too contracted escape from below the baffle and a restricted flow from the eduction pipe; (2) inefficient packing of the tube glands, or shrinkage of the

packing when dry—for this reason the water should pass through the tubes, and the condenser should not be drained unnecessarily on the circulating side; and (3) too great clearance in the holes in the diaphragm plate.

If a sufficiency of lime is mixed with the feed water, the effect of acid corrosion is practically overcome. The other corrosive agent is the air or oxygen set free from the condensed steam or water, which in its nascent state has a great affinity for metals. If the circulating inlet is below the condenser, and the discharge gradually rises to the outlet, the heating of the circulating water, obtained by the condensation of the steam, evolves the oxygen and air, which pass away through the outlet, and little corrosion takes place; consequently, when fitted in this way, after shutting off steam, the inlet should be closed to prevent the entry of air, and the outlet left open to allow any air to escape. It is now customary to fit air outlet pipes near the highest points of the circulating water spaces to allow the air to escape; this practice is of undoubted advantage in preventing corrosion. On the steam side the same reasoning holds good; but as the air may have access through the eduction pipe as well as the air-pump suction, these should be kept closed when the condenser is not in use, and no air allowed to enter. It is impossible to drain the condenser on the steam side without leaving moisture on the tube surfaces, and the exclusion of air is the best practice.

Air Pump.—Originally the cylinder itself was cooled after each stroke, to produce a vacuum; but it was soon evident that a separate vessel for this purpose would be advantageous, and James Watt introduced the condenser. In this way the steam cylinder is always kept as hot as possible, and the separate chamber in connection with the cylinder, called the *condenser*, is kept as cold as possible. Hence it follows that directly the steam enters the cylinder it is ready to act more effectively; and directly communication is opened with the condenser the exhaust steam makes its escape, assisting in its own withdrawal by producing a partial vacuum from condensation in the condenser. At this point another difficulty arises—the condensed water and uncondensable air and gases would soon fill the condenser, unless continuously removed.

The air pump is used for this purpose, and its function is therefore to pump the condensed steam and uncondensable vapours and gases from the condenser, and at the same time to decrease

the pressure in the condenser by continually pumping out its contents. The condenser and air pump are mutually dependent on each other for reducing the temperature and pressure of the exhaust respectively. The effect of this reduction of temperature and pressure is to reduce the back pressure on the piston (only the L.P. piston directly in a stage-expansion engine), and also to increase the possible ratio of expansion, thus producing economy of steam and fuel.

The air pump discharges its contents into either the hot well, the grease filter, or the feed tank; from one or all of these the air is allowed to escape through vent pipes fitted for the purpose. From the feed tank the boiler feed pumps take their suction, and thus the same water is used over and over again. With a surface condenser, and where the packing glands of the cylinders and rods are well fitted, the amount of air which actually enters the condenser is small, and thus the air pump could be of small size; but provision must be made for emergencies, and, to prevent overcharging, the air pump is generally a large fitting.

Until recently the air pump was usually a single-acting bucket pump, generally worked off the main engines either by levers from the piston-rod end or by direct connection with one of the pistons; the former method is preferable. Double-acting pumps are now frequently used, and save space and weight. Recently, for naval purposes, and also for some of the better class of mercantile marine, the air pump is worked by an independent engine. Two air pumps are now fitted for each main engine in the larger vessels, each of which, being independent of the other, can be worked or repaired as required.

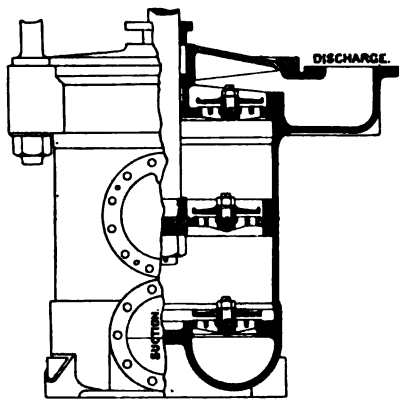


FIG. 199.—Single-acting Air Pump.

In Fig. 199 a vertical air pump is shown. There are in this single-acting type three sets of valves, one each for *delivery*, *bucket*, and *suction*. The valves are preferably made of gun metal; but india-rubber is sometimes used, and it is then specially prepared and vulcanised. The bucket, or plunger, is fitted with packing, to

make it a working fit in the barrel, and for large engines a junk ring is fitted to it for fitting or renewing the packing. The packing is preferably made up of two or more vulcanite rings, generally uncut or solid. Hemp is also used, and for small engines is efficient; and in some cases a broad ring, made of gun metal, is used, fitted with a tongue piece and water grooves. Solid plungers are occasionally used for air pumps, and the fitting of water grooves adds considerably to their efficiency.

Metallic Valves.—A type of valve, known as Kinghorn's, is very frequently fitted to air pumps, and is generally most efficient and durable. The arrangement is shown in Fig. 200, and consists of three gun-metal plates, each about $\frac{1}{16}$-inch thick; the two lower plates are perforated with holes, in circles *not* immediately over each other. It is usual to fit the discharge valves with a light brass spring, to return the valves to their seats after lifting. The lift allowed for moderate-sized valves is about $\frac{3}{8}$-inch. Other methods of valve fittings are numerous. One consists of a series of concentric corrugations on a metal plate, and is also frequently used. Within the limit of sufficient strength and durability, the lighter the suction valve can be made, the greater the possible efficiency of the air pump. The weight of the suction valve practically limits the vacuum attainable, for it is evident that, to be capable of lifting the valve, the pressure inside the condenser must be sufficiently in excess of that above the suction valve.

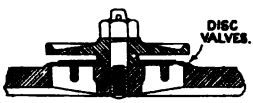


FIG. 200.—Kinghorn's Metallic Valves.

The efficiency of the air pump is a most important factor in the efficiency of the engine, and every opportunity should be taken to insure its proper condition by examining and refitting the valves and packing. The pump should be so placed that, under ordinary working conditions, the suction valves are at a slightly lower level than the bottom of the condenser.

It is usual to make all parts of the air pump, its working barrel, plunger, valve seatings, and guards, of gun metal, except the plunger rod, which is made of rolled naval brass.

Air Vessel.—With a rapidly reciprocating pump which, like the air pump, is subjected to greatly varying load, the impact of discharge of an almost incompressible fluid, such as water, produces great shock on the discharge pipes and working parts of the pump.

By using an air vessel, connected with the discharge, and in which some air is always imprisoned, the force of impact is reduced by the elastic cushion thus formed.

A type of air vessel is shown in Fig. 201. It is fitted on the air-pump discharge, and the more or less solid water has a clear passage through it. Once inside the upper part of the pipe, the air pressure contained in the vessel supports the weight of water above it, and serves to cushion the impact of the next discharge. The upper internal pipe is preferably made of zinc or cast iron, which, itself corroding, protects the gun-metal fitting and copper discharge pipe from corrosion. The air vessel should be fitted immediately above the discharge valves.

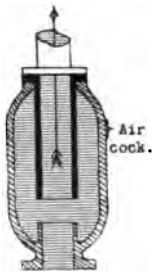


FIG. 201.—Air Vessel.

Edwards' Air Pump.—This pump differs from the ordinary single-acting bucket pump, as previously described, in the form and shape of the plunger, and the omission of both bucket valves and suction valves. The plunger is of the solid type, but is of hollow construction. The upper face is flat, and the lower part is made in the form of a cone, as shown in Fig. 202. The plunger works in a gun-metal barrel, *A*, which is surrounded by an exterior casing, *B*, in uninterrupted connection with the lower part of the condenser through the suction pipe. The casing at its lower part is similarly shaped to the plunger, and the plunger, when at the bottom of its stroke, nearly touches the casing.

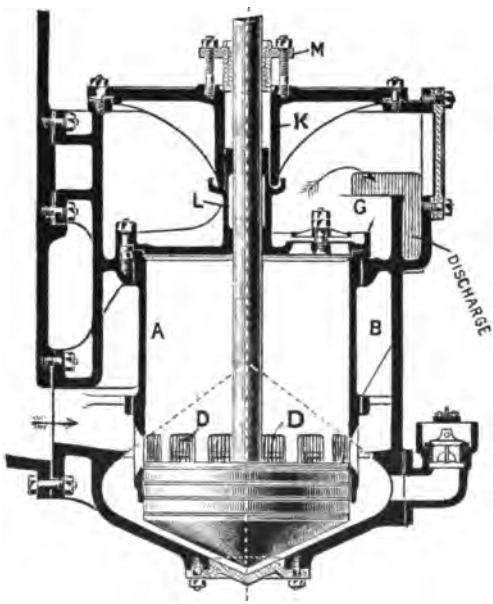


FIG. 202.—Edwards' Air Pump. (One view only.)

The void left by the plunger on the up-stroke produces a flow of the condensed water and uncondensable vapours into the bottom

of the casing; and when the plunger descends it compresses the contents, and forces them through holes, *D*, in the working barrel into the space above the plunger. The casing must be suitably shaped to effect this purpose. The inducing action thus set up tends to continue the flow into the casing from the condenser. The plunger on the up-stroke catches the contents above it, closing the holes *D*, and nearly the whole of the contents above this level are discharged during the up-stroke through the delivery valves. On the down-stroke the valves are closed, and an almost complete vacuum is formed above the plunger. As soon as the holes begin to

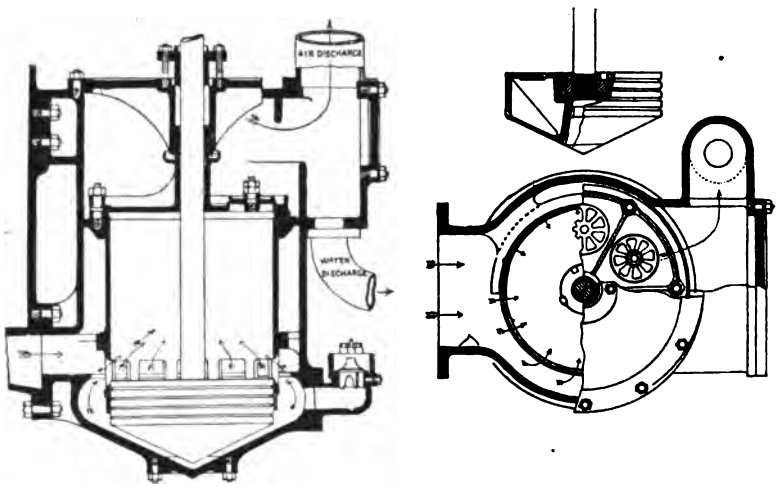


FIG. 202A.—Edwards' Air Pump. (Two views.)

open, the contents of the condenser flow into the barrel, and the action is continued by the forcing and inducing action of the plunger and the directing action of the casing. The flow from the condenser into the casing is almost continuous, and adds to the efficiency of the pump.

The plunger is not always fitted with packing rings, but water grooves are cut circumferentially in the working face, as shown. The rod is fixed to the plunger by a coned end, drawn up to its seating by a sunk nut on the upper side of the plunger; and the nut is secured by set screws, partly fitted in the plunger and partly in the nut. The rod works through a long sleeve, *L*, which is water-packed for some part of its length, while the upper part, *K*, is

fitted with a stuffing box and gland, *M*, cupped at its top end round the rod for moistening the packing.

The top end of the working barrel is fitted with a cover, in which the valves are seated. The cover is formed with a lip, *G*, so that the valves are always water-packed when the pump is working, and air leakage into the barrel is thus prevented. The valves are accessible when the pump is working, but, of course, there is some inconvenience and loss of water.

The lower part of the casing is fitted with a non-return relief valve, to prevent damage from any excess of pressure.

Edwards' air pump can be driven by any of the usual means, either from the main engines direct, or by independent engines. The efficiency obtained is generally greater than with the ordinary single-acting pump, and it is fitted in a large number of vessels of the mercantile marine, and in many warships, for both main and auxiliary purposes.

Independent Air Pumps.—Many vessels of large power are now fitted with air pumps worked by independent engines, and, following the same lines as in earlier days, when the feed pump was detached from the main engine, there appears to be some gain and advantage obtained by relieving the main engine of all duty except that of revolving the propeller shaft.

The advantages claimed are :—

1. A vacuum can be obtained at all times, and its production allows the engines to be kept properly drained and ready for reversing or manœuvring.

2. The main engine is relieved from a large amount of work when starting.

3. The adoption of two main condensers, with independent pumps, allows the auxiliary condenser to be done away with, thus saving some weight and a lot of cumbersome complication.

4. The independent pump readily accommodates itself to any extra work thrown on it.

In the Weir arrangement (Fig. 203), as fitted in the ss. *Carmania*, there are two pumps side by side, driven by two steam cylinders side by side above them, and the same steam valve serves for both cylinders. The valve gear is worked by levers from the piston-rod ends (the usual arrangement of Weir pumps and valve gear), but in this case in connection with a rocking beam. A

rocking beam, *AB*, with short connecting rods *C* and *D*, joins up the steam piston rods *E* and *F*, so that they reciprocate in opposite

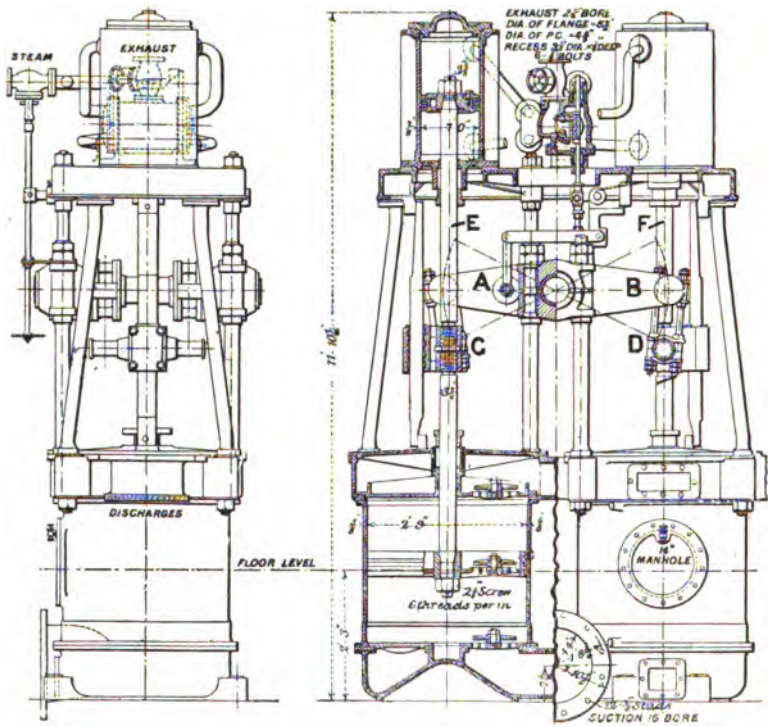


FIG. 203.—Independent Air Pumps.

directions, one being at the top and the other at the bottom at the same instant.

Dry Air Pumps.—For turbine installations an augmenting apparatus is commonly used in conjunction with the ordinary air pumps. For reasons which are more fully explained later, a high vacuum is more advantageous with turbines than with reciprocating engines, and its attainment in some cases increases the efficiency of the steam by 5 to 8 per cent. In the *Mauretania* two double-cylinder dry air pumps (Fig. 203A) are fitted of the Weir single-acting type, with an annular opening in the pump barrel and without suction valves. The pumps are cooled by a small supply of circulating water which passes through the air chamber. The dry air pumps are smaller than the wet (or ordinary) air pumps, and

the stroke is shorter, but the plunger displacement per minute is about the same for each type.

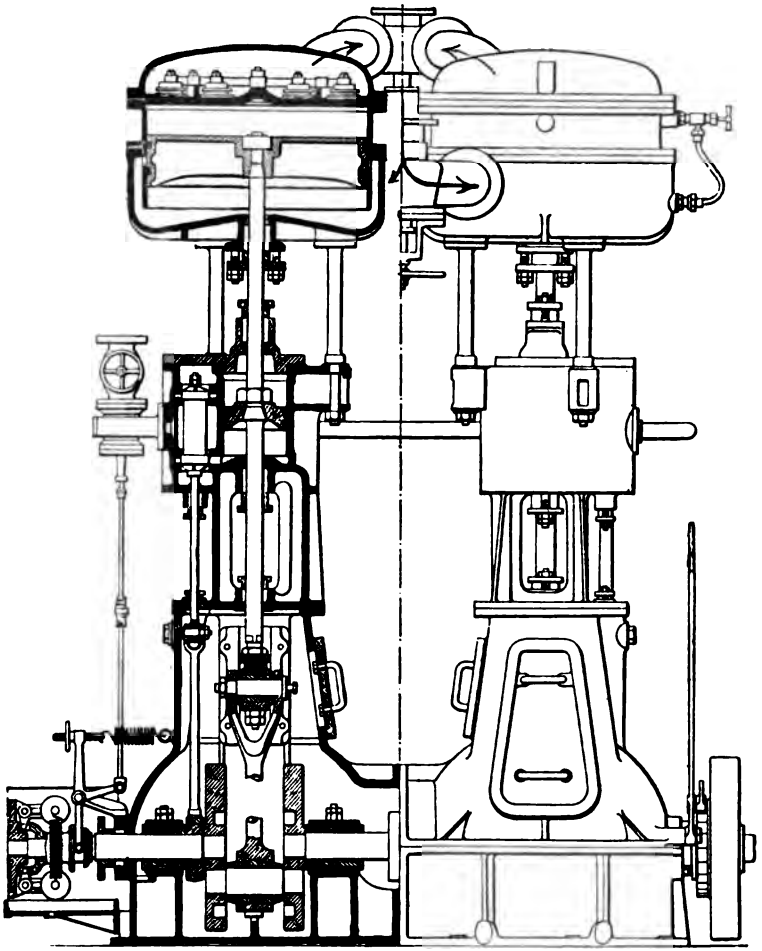


FIG. 203A.—Dry Air Pumps (Weir).

Parsons' Vacuum Augmentor.—In many cases with turbine installations, and in lieu of dry air pumps, a Parsons' vacuum augmentor is fitted. By this means a very high vacuum, frequently 29 inches or over 96 per cent of a perfect vacuum, has been obtained on service. As shown in Fig. 204, the air pump is placed about 3 feet below the bottom of the condenser, which

should preferably be sloped so as to drain naturally into the air-pump suction *A*. From another position *B*, somewhat above the level of *A*, a suction pipe, *C*, leads into an auxiliary (or augmentor) condenser containing about one-twentieth the cooling surface of the main condenser. In a contracted portion of the pipe *C* is fitted a small steam nozzle of the De Laval type which, acting as an exhauster, sucks air and vapour from the main condenser and delivers them through the augmentor condenser into the air-pump suction pipe *D*. A water trap, or seal, formed in the suction pipe *D*, prevents the return of the air and vapour to the condenser. For the efficient working of the augmentor, it is necessary that its

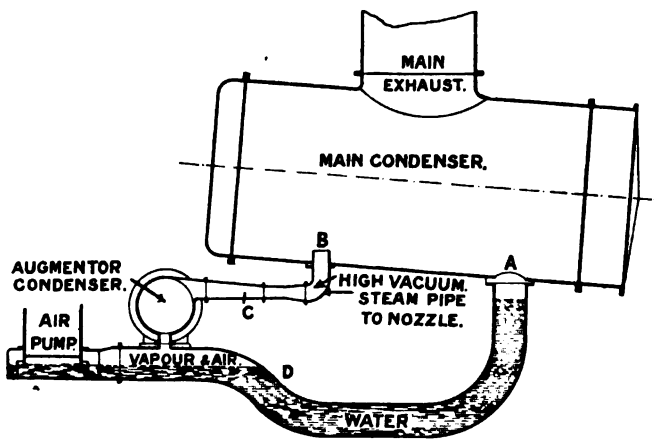


FIG. 204.—Parsons' Vacuum Augmentor.

suction *B* should be above the main suction opening *A*; that the steam nozzle should be proportioned to its work and discharge directly in line with its delivery pipe (and preferably downwards); and that the condenser, nozzle chamber *C*, and condenser suction pipes should be naturally self-draining, with the exception of the water trap mentioned above.

While fully admitting the great economical advantage of high vacua for turbine engines, and taking into account the extra steam consumption of $1\frac{1}{2}$ per cent for the steam jet, and the increased cost of the augmentor condenser and the weight and space involved, it is extremely doubtful whether either dry air pumps or vacuum augmentors really repay the expenditure on their installation and working. With well-fitted naval reciprocating engines a vacuum of

28 inches (temperature 100° F.) is by no means uncommon, and there is little doubt that with turbine engines, where the leakages should be less, a vacuum of 28½ inches (temperature 90° F.) should be obtainable by fitting a more efficient pump (Edwards', for instance) and by paying particular attention to the relative positions of the various pipes and fittings. To reduce the temperature of the feed water below 90° F. is generally attended by a higher fuel consumption in reciprocating engine practice, and a feed water temperature of over 100° F., although only theoretically corresponding to a vacuum of 28 inches, is commonly the object aimed at.

Circulating Pumps.—The pumps used for circulating the cooling water through the condensers are nearly always of the

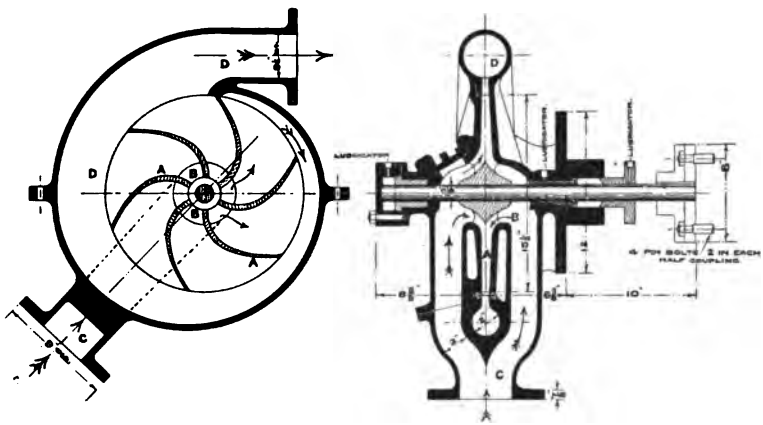


FIG. 205.—Centrifugal Pump.

centrifugal type. The general arrangement of the inlet and outlet pipes is shown in Fig. 207, which also shows the position of the pump and the independent engine used to rotate it.

The *impeller*, or runner, which is shown in shaded section in Fig. 205, is fitted with a series of vanes, *A*, curved in such a way that the centrifugal action imparts a fairly high velocity to the water which is drawn in at *C* to the middle part *B* of the impeller, and thrown off at the periphery or outer circumference *D*. The arrow heads show the direction of rotation and of the course of the water through the pump. There are no valves in the pump to get out of order; and as the pump is always situated some distance below the outer water-level, the pump chamber, suction, and inlet pipe are always kept charged. The pressure the pump has to exert

is very small, simply sufficient to overcome the internal friction of the pipes, tubes, and passages through which the water passes, and consequently a very large amount of water can be circulated through the condensers, with a high efficiency.

Each pump is generally driven by a separate engine, and a pair of pumps are usually fitted to each main engine. One pump can supply sufficient circulating water for all but the very highest powers, so that the circulating system is practically in duplicate. The shaft shown in the sketch runs in two *lignum vitæ* bearings. The bearing next the engine is also fitted with a stuffing box and gland, to prevent the water leaking through; the outer end of the shaft and its bearing are generally fitted with a cap, which encloses them. Both these bearings are cooled with a water service, and in the later types a pipe is led from the top of the pump chamber to the top of each bearing, to maintain a circulation through the *lignum vitæ* strips.

All parts of the pump and casing in contact with the sea-water are made of gun metal or bronze, except the *lignum vitæ* bearings already mentioned. A cock is fitted at the highest part of the pump chamber for letting out any air which may accumulate, and also for testing the efficiency of the pump by observing the height to which the water is thrown. Any decrease in the usual height generally points to some choking of either the inlet valve or pipe.

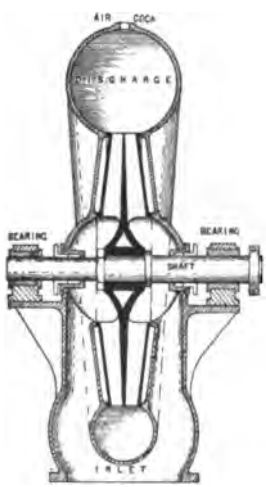


FIG. 206.—Centrifugal Pump.

In Fig. 206 a somewhat different arrangement of the circulating-pump bearings is shown. The inlet water enters from both sides of the impeller, and the pumps indicated in Fig. 207 are made in this way. The impeller shaft is fitted with outside bearings of gun metal, with white-metal rubbing surfaces, which are lubricated in the usual way. Stuffing boxes and glands are fitted to keep the shaft water-tight where it passes through the pump casing. This arrangement is particularly useful when navigating in shallow waters, such as the Suez Canal, as the sand and grit carried in with the circulating water is thus excluded from the bearings; it is adopted in the *Mauretania*. In this type of pump it will be

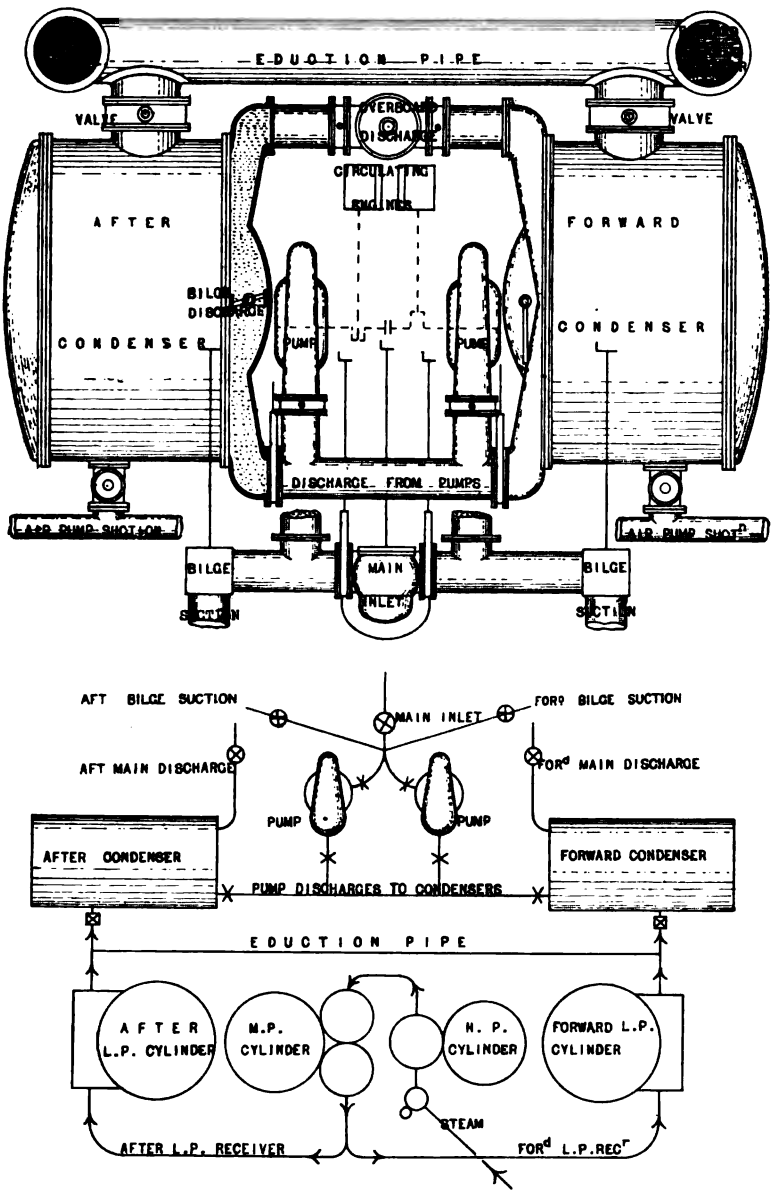


FIG. 207.—Arrangement of Condensers and Circulating Pumps.

noticed that the ends of the impelling blades terminate at the inner part of the outlet tube, and for the same diameter of casing a smaller diameter of impeller is used than in the previously described pump.

Circulating Bilge Suction.—To each pump a connection is fitted to draw water, if required, from the engine-room bilge. If required for pumping out the stokeholds, these can be drained by isolated independent pipes to either engine-room bilge, and then be pumped out from there. The arrangement is shown in Figs. 193 and 207, which also show the rods for the working of the valves. The handles or wheels should be accessible when the engine room is flooded, while still leaving the centrifugal pumping engine free to be worked. Unless particular attention is paid to the design of the suction valve so that it can lift easily and accommodate itself to the varying fluctuations of the pump and the run of water to the suction, no great reliance can be placed on the centrifugal circulating pumps to remove much water from the bilge. Apart from the inefficiency of the pump itself as a suction pump, there are numerous obstacles in the engine room, such as frames and pipes, which all prevent a good flow to the suction, and render its efficacy doubtful. In recently constructed vessels, larger holes in the framing are made to allow a better flow to the suctions.

A large modern warship is generally fitted with four circulating pumps, two to each main engine, and their united capacity is from about 6000 to 10,000 tons per hour. This apparently great quantity is very small compared with the amount of water which will enter a hole of very moderate size at a depth of about 20 or 25 feet below the water-line. In emergency every effort should be made to reduce the flow of water into the ship, and then a possibility exists for keeping the reduced leakage within safe limits. The pumps are generally practically tested in the ship with a reduced steam pressure and number of revolutions.

Heat rejected in the Condenser.—The exhaust steam enters the condenser at about 3 to 4 lb. absolute pressure per square inch, corresponding to about a temperature of 150°. This steam in practice contains by weight from 20 to 25 per cent of moisture in suspension. If considered as dry saturated steam, then every 1 lb. weight of steam carries with it into the condenser about 1010 b.t. units in the form of latent heat, and, in addition, the cooling water reduces the temperature to about 100°, the temperature of the feed water. Therefore the heat carried away by the circulating

water, or the heat rejected in the condenser, is for each 1 lb. of steam used about—

$$1010 + 50 = 1060 \text{ b.t. units.}$$

For a modern battleship or cruiser, about 15 lb. weight of steam is used per I.H.P. for the main engines, and about 2 lb. for the auxiliary engines, per hour. Taking the amount at 15 lb., the heat rejected is (considering the exhaust to be dry saturated)—

$$1060 \times 15 = 15{,}900 \text{ b.t. units per I.H.P. per hour.}$$

Quantity of Cooling Water required.—The heat rejected in the condenser decides the quantity of water required to cool the exhaust steam and condense it into water. A rise of about 25° in temperature is allowed for the circulating water, and assuming that each 1 lb. weight of steam conveys 1060 units of heat into the condenser, the quantity of cooling water required is—

$\frac{1060}{25} = 42{\cdot}4$ lb. per 1 lb. weight of steam used by the engine.

If n lb. of steam are used per I.H.P. per hour, then the quantity of cooling water is—

$$= n \times 42{\cdot}4 \times \text{I.H.P. per hour (weight in lb.),}$$

and the circulating pumps must be capable of forcing about this amount through the condensers per hour.

In the Tropics and narrow waters the temperature of the sea is from 80° to 85°, and if a lower condenser temperature is required, it is obvious that the quantity above must be increased. With a temperature of discharge of about 105°, a vacuum of about 28 inches is possible; and in practice the quantity stated above is considered more than sufficient, and about 40 lb. is usually allowed for purposes of naval design of piston engines; but for turbines a much greater quantity is considered necessary.

NOTES, BEARING ON DESIGN OF CONDENSERS, ETC.

FOR NAVAL RECIPROCATING ENGINES

23,500 I.H.P.

Condensers built of gun metal throughout, with end covers and sight-hole doors (12 in. diameter).

Tubes of solid drawn brass tested to 300 lb. per square inch internal pressure.

10 ft. long × § in. diameter and 0·048 in. thick.

Cooling surface of tubes, square feet—
7008 for 1 condenser.

Total 28,032 for 4 condensers
= 1·19 sq. ft. per 1 I.H.P.
(not including I.H.P. of auxiliaries which exhaust into main condensers).

Air pumps—

2 twin independent.

Circulating pumps—

4 sets.

Fitted with 4 bilge suctions for use in emergency.

Water discharged = 40 × weight of feed water.

FOR TURBINES, MERCANTILE PRACTICE

20,000 shaft H.P.

Condensers built of steel plates, with brass end covers and doors.

Tubes of solid drawn brass, tinned inside and outside.

Cooling surface of tubes, square feet—
27,030 for 2 main condensers.
5,406 for auxiliary condenser.

Total 32,436 sq. ft.
= 1·62 sq. ft. per 1 S.H.P.

Air pumps—

2 twin independent (Weir), 33 in. diameter and 21 in. stroke.

2 double-acting (dry) pumps, 20 in. diameter and 7 in. stroke.

Circulating pumps—

2 sets with gun-metal impellers 4 ft. 6 ins. diameter, driven by steam cylinders 14 in. diameter and 12 in. stroke; suction and discharge pipes, 23 in. diameter. Water discharged = 50 to 60 × weight of feed water.

CHAPTER XVIII

STEAM, EXHAUST, AND DRAIN SYSTEMS

Arrangement of Main Steam Pipes.—There are four systems to be found in warships of various dates, and they may be considered in turn.

1. The arrangement fitted, about 1880, for a twin-screw ship was generally adopted for ships with two boiler rooms. The steam from the forward, or starboard, boilers was led aft to the engines by one pipe, and the steam from the after, or port, boilers was led aft by another pipe; the two pipes were connected with each other through a cross-connecting pipe at the forward end of the engine room, with a shut-off valve on the bulkhead between the two engine rooms. A self-closing valve was fitted at the end of each pipe at the athwart-ship bulkhead; by this means either engine can be worked from any set of boilers, or from one boiler only. Such an arrangement is frequently fitted in the mercantile marine. In a few cases another cross-connection pipe and valve is fitted at the forward end of the foremost boiler room.

2. As the power, necessary size of pipes, and number of boilers increased, it was found advisable to fit a separate pipe for each group, generally confined to one boiler room, and to lead it separately aft to the engine-room bulkhead, where a self-closing valve was fitted. This arrangement is common in the cylindrical-boilered ships of 1886 to 1897.

3. The introduction of the water-tube boiler, and a large increase in the power and number of boilers fitted to each ship, led to a still further development, which is shown in Fig. 208. The boilers are grouped conveniently into sections. The steam from each boiler is led into a sub-section box, which acts as a collector, and thence by a single pipe to the section box; a separate pipe

leads from each section box to the engine-room bulkhead, where a self-closing valve is fitted at the end of each pipe.

In the figure the main steam pipes are shown in full lines, while the auxiliary system is shown dotted. The numbered rectangles are the positions of the boilers as fitted in some of the large County cruisers, and the thick lines are the boundary bulkheads of the boiler rooms and engine rooms. Three boiler rooms are shown with two stokeholds in each; but independently of this division, the boilers are divided into four groups, each of which is connected by a separate pipe and bulkhead valve with the cross-connecting pipe. For example, the steam from Nos. 1 and 2 boilers is led by separate pipes into a sub-section box, thence one pipe leads into the section box. By the arrangement shown in the enlarged views of the section box, the steam can be conducted into either the main or auxiliary system. The main lead passes from the section box through the other boiler rooms, but is not connected with the steam from the other groups until it reaches the engine-room bulkhead, where it can be connected with the cross-connecting pipe through the self-closing valve *B*. From here it can be led into

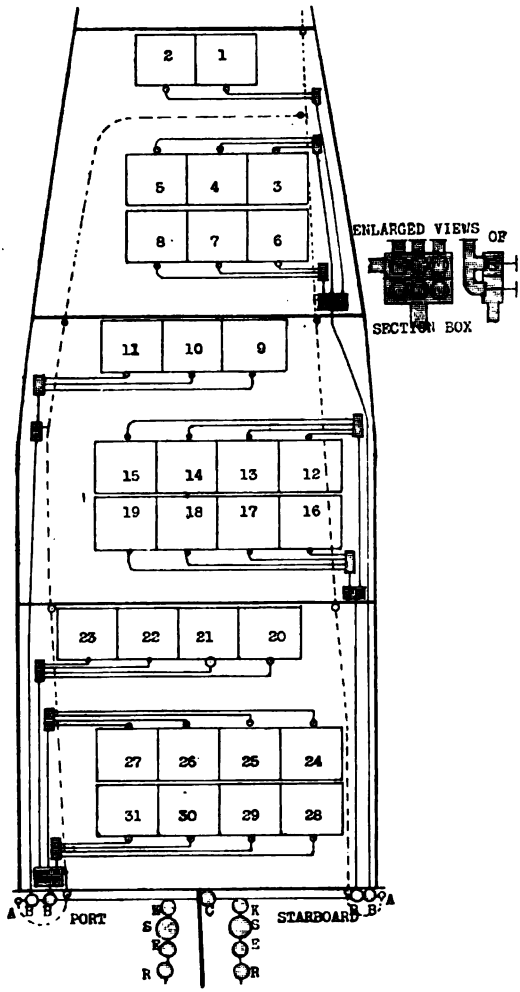


FIG. 208.—Main Steam System (County class).

the other engine room through the valve *C*, which can be closed from either engine room; or it can be led into the engine through the main shut-off valve *K*, the separator *S*, the emergency valve *E*, and then through either the main regulating valve or the manœuvring valve *R* to the engine; or, by opening the valve *A*, steam can be led into the auxiliary system, but this is not necessarily opened without a particular reason. A connection is also made at *B*, through valves and pipes, for working the reversing, steering, and one main circulating engine with steam from the main supply. The other circulating engine is connected with the auxiliary steam system for use in harbour when the main steam supply is not opened.

4. Since the introduction of steel and wrought-iron steam pipes, which can be safely used and made in larger sizes than the previous copper pipes with braised joints, there is a tendency to simplify the system. Now a single pipe on each side of the ship is used, if possible, as shown in Fig. 209; but as the boilers now almost invariably face fore and aft, the steam is collected by cross-connecting pipes, each of which is connected with one of the main leads by a self-closing valve, *D*.

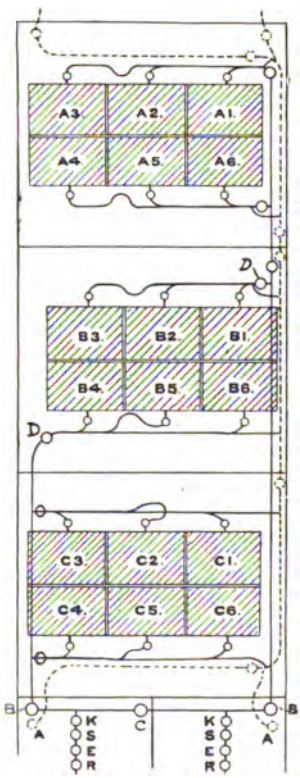


FIG. 209.—Main Steam System (Battleship).

Auxiliary Steam System. — The auxiliary steam pipe runs through the engine rooms and boiler rooms, as shown in Figs. 208, 209, 210, and 212. One or two branches lead to the auxiliary engines abaft the engine rooms, and there are generally one or two branches (Fig. 210) leading to the forward part of the ship for the same purpose. From the main circuit short branches lead to each separate engine, and branches lead the exhaust steam into the exhaust pipe, which generally closely follows the run of the steam service.

The auxiliary steam pipe is supplied with steam, either directly

from the boiler through a separate auxiliary steam valve, or from the main system, as shown in Figs. 208 and 209. An entirely separate system of valves and piping for the auxiliary steam service is preferable, because, in case of accident to the main steam pipe, the auxiliary can then be brought into use; and there are other practical advantages.

Auxiliary Exhaust System.—The auxiliary exhaust service

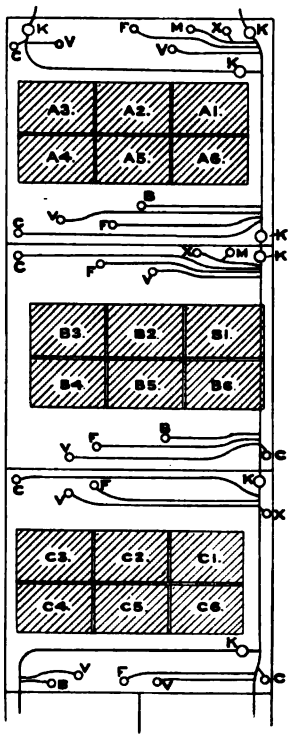


FIG. 210.—Auxiliary Steam System.

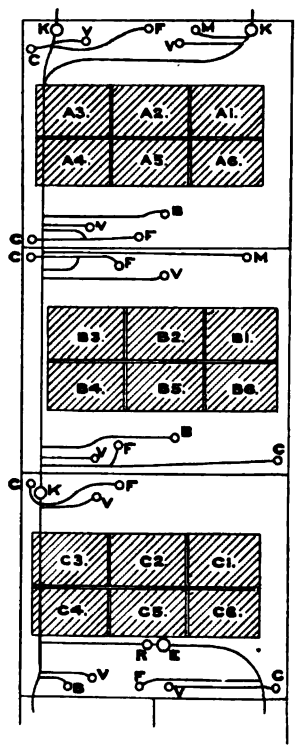


FIG. 211.—Auxiliary Exhaust System.

also runs through the boiler rooms and engine rooms, as shown in Figs. 211 and 212, generally following the run of the steam service. Branches lead from the main run of pipe to each auxiliary engine. The branches leading to the fore part of the ship, before the foremost boiler room, can be shut off by valves, *K*, fitted near the bulkhead.

The steam and exhaust branches, shown in Figs. 210 and 211, lead to the following:—

X, Ash-ejector; one to each boiler room.

B, Air blower; one in each boiler room, supplying air to the fires through the gas mixers.

C, Oil fuel pumps, for supplying oil to burners.

E, Atmospheric exhaust pipe, leading to the outer air outside the funnel, and used to carry away any excess pressure escaping through the relief valve *R*.

F, Feed engines. The main feed engine is connected directly with the main steam pipe.

K, Bulkhead stop valve at forward end of pipe, and shut-off valves in pipes.

M, Fire and bilge pumps.

R, Relief valve, allowing excess pressure to escape to atmospheric exhaust pipe.

V, Ventilating fan engines for supplying air to the furnaces through the ashpits.

In the engine room the auxiliary exhaust pipe runs aft to the condenser, and a cross pipe with a bulkhead valve connects the condenser with the other engine room. When no auxiliary condenser is fitted, the cross-connecting pipe joins up the two main eduction pipes through the bulkhead valve. When the main engines are not under steam, one of the main condensers, a circulating pump (connected with the auxiliary steam service), and an independent air pump, are used. The last is not always necessary, especially in small vessels, as the condenser can sometimes be drained through a non-return valve into the feed tank; but the use of the air pump is preferable, as the water can then be pumped into the grease extractor before its return to the boiler.

Closed Exhaust System.—Since 1897 many warships have been fitted with an economising device known as the closed exhaust system, and the arrangement of auxiliary exhaust pipes in Fig. 212 can be used in this way. The auxiliary exhaust is throttled at its connection *T* with the condenser, and a pressure of about 27 lb. per square inch is maintained in the exhaust pipes by means of a spring-loaded relief valve, *R*. Steam from this source can be used—

(*a*) In the L.P. cylinders, by passing through a relief valve in connection with the L.P. receiver, set at 25 lb. pressure, and a shut-off valve, *N*, workable from the starting platform.

(*b*) In the evaporators, by passing through a water collector, *M*, shut-off valves, *E*, and a spring-loaded valve, *L*, set at about 27 lb.

pressure, into the evaporator heating coils. After doing work in the evaporator, in which the steam is condensed by boiling the water surrounding the coils, the condensed steam is passed into the condenser through the valve *S*.

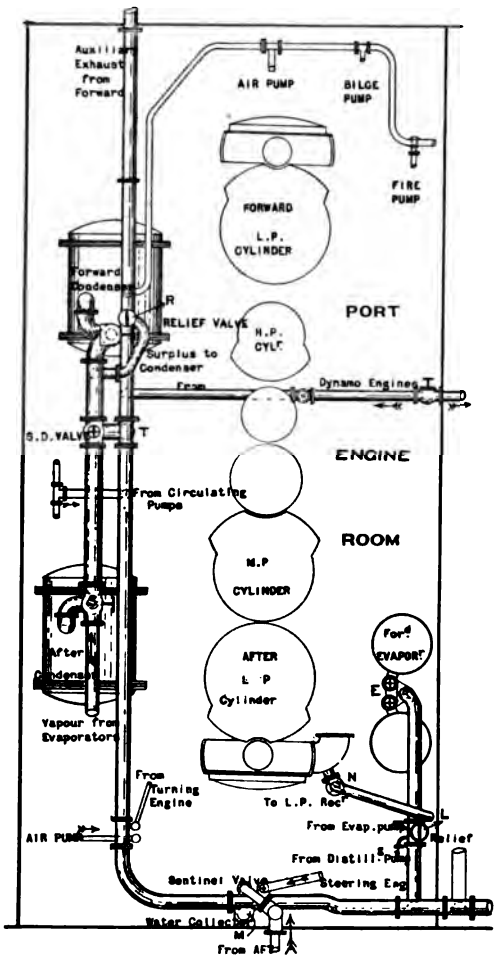


FIG. 212.—Closed (auxiliary) Exhaust System.

By using the closed exhaust system a large quantity of heat, otherwise wasted, is converted into useful work, and there is a considerable gain in economy. The back pressure on all the auxiliary engines in connection with the system is considerably augmented, however; and in some cases, where the system was fitted after the ships were originally fitted out, some difficulty has been experienced with the feed engines, owing to the consequent reduction of the mean pressure on the pistons, which under the new conditions were too small for their work. In the more recently designed ships better results have been obtained, and the practical gain is definitely marked.

The practical objection to the use of the closed exhaust is the water loss caused by increased leakage from the numerous piston, slide, and valve-rod glands throughout the auxiliary system. When a small vacuum can be maintained in the exhaust pipe, nearly all this leakage is prevented; the feed pumps and other engines are better drained and more efficient; and the evaporators

are less used, with a consequent saving in steam and fuel. Some weight and complication is added by fitting the closed exhaust; but on the whole it has justified its existence, and very economical results have been obtained—amounting to a production of 10 to 12 tons of fresh water for each 1 ton of coal burnt. (See also Chapter XX.)

Expansion Joints on Main Steam Pipes.—The pipes conducting the steam from the boilers to the engines are long, sometimes about 100 feet, and the variation in length, due to their expansion when hot, is considerable. For ordinary temperatures and high pressures about $2\frac{1}{4}$ inches must be allowed for expansion in every 100 feet of length of steam pipe. The expansion may be allowed for in two ways—by making the pipes with bends of sufficient curvature, or by fitting expansion glands. The former method is generally used for pipes of small diameter, and the latter for pipes of large diameter.

Each alteration of direction of the path of fluid pressure through a pipe produces a consequent end thrust, which must be considered in connection with the allowance for expansion. For example, consider a pipe, with a right-angled bend, containing steam of 250 lb. pressure per square inch, and about 15 inches in diameter; then the total pressure tending to force the pipe away from its end connections is equal to the pressure × area of the pipe = about 20 tons, for the figures stated above.

The radius of curvature may be as great as possible, but it does not reduce the unbalanced end pressure or force, tending to produce rupture at the ends of the pipe. There is some mechanical difficulty in making a curved pipe of uniform strength and thickness suitable for a varying pressure and temperature, and the general practice is now to construct the necessary bends in short lengths of cast steel, while the straight pipes are made of steel or wrought iron, welded at the laps, and covered with a riveted butt strap.

The necessary expansion is allowed through expansion glands, and the end pressure is safeguarded by stays fitted between the flanges of the parts of the pipe. In Fig. 213 the end of one part of the pipe is made with a stuffing box, *A*, into which the opposite end, *B*, of the pipe projects, and is made steam-tight by the packing gland *C*, generally packed with alternate layers of asbestos and elastic core packing. The unbalanced end pressure is taken on the

guard stays, which are fitted through lugs behind the flanges or made in one with them. The check nuts hold one end of each stay fast, while the other can move through the holes within the limit of the bolt head. The check nuts are pinned through the stay, to prevent their moving or maladjustment after removal.

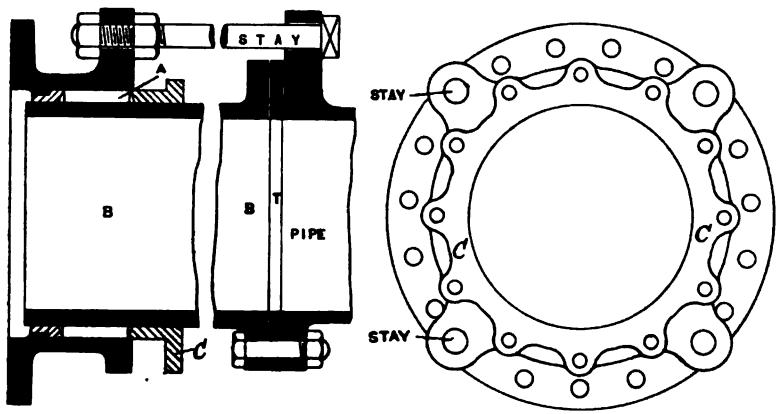


FIG. 213.—Pipe Expansion Gland.

The ring *T* is a distance piece, and when necessary to test the pipes by water pressure, it can be removed and a blank flange substituted. Arrangements are made in a similar manner for the various pipes between the boilers and the bulkhead valves, the next length to the regulating valve, the various receiver and eduction pipes, and parts of the auxiliary steam service.

Bulkhead Valve.—The separate steam pipes leading from the boilers to the engines are controlled by the bulkhead valves (*B*, in Fig. 208), which are of the *self-closing* type, and similar to the boiler steam valves. A type of bulkhead valve is shown in Fig. 93.

Intermediate Shut-off Valve.—After passing through the bulkhead valve, the steam enters the cross-connecting pipe, and the supply to each engine is governed by the intermediate valve. This valve is of the ordinary screw-down type, and opens against the steam pressure; consequently, when closed, it is generally steam-tight. A small pass valve and pipe are generally fitted in connection with large valves, to equalise the pressure on opposite sides of the large valve, so that the latter can be opened easily. Gearing is fitted in connection with the valve, to work it from outside the

engine room, and it should be accessible without difficulty from the engine-room platform.

Separator.—The steam next passes through the separator, the object of which is to extract any water or moisture from the steam before its arrival at the engine. The simplest form of separator consists of a vertical cylindrical chamber, with a diaphragm or dash plate separating the upper part of it. The mixed steam and water enters by the inlet, and is directed downwards by the dash plate; the water is left in the bottom of the chamber, and the lighter steam escapes upwards and passes into the outlet pipe, on its way to the engine. A water gauge is fitted at the lower part of the chamber, and any accumulation of water can be blown, through a blow-down valve, into the condenser, or into the feed tank in a few instances. The separator is a very useful fitting, and many accidents from priming and other causes have been prevented by its use.

The separator, as described above, although very useful in dealing with large quantities of water, is not very efficient when dealing with only a moderate percentage, because the particles of moisture, deposited on the separator plate, drain down to the bottom edge, only to be again picked up by the steam in its passage to the outlet.

An improved form of separator is the *Lancaster central-pipe steam dryer*, shown in Fig. 214. The arrow heads show the course of the steam. Entering through the inlet pipe, the steam is deflected by a web, *A*, made outside the central pipe *B*, and passes spirally downward outside the pipe; thence, turning upwards, it passes through the interior of the pipe to the outlet. Any water deposited on the exterior of the central pipe drains into a cupped lip, *D*, at the lower end of the pipe; and a drain pipe, *E*, leads the water into the bottom of the dryer. Particles of water are also thrown off, by the steam in its passage, by centrifugal action. These are collected by vertical ribs, *C*, made in the inner face of the outer containing vessel, whence the water drains downwards into the bottom of the dryer. The drain water is automatically removed through the connection at *F*, by a steam drain trap. At *G*, a cock or valve is fitted for blowing down and other purposes.

As far as possible, the steam pipes are made *self-draining*. There should be a gradual fall in level from the boiler steam valves to the separator, whence the water can be blown out.

Another form of steam dryer is shown in Fig. 215, in which a

baffle plate is perforated with a number of small holes to allow the separated water to pass into the drain vessel below.

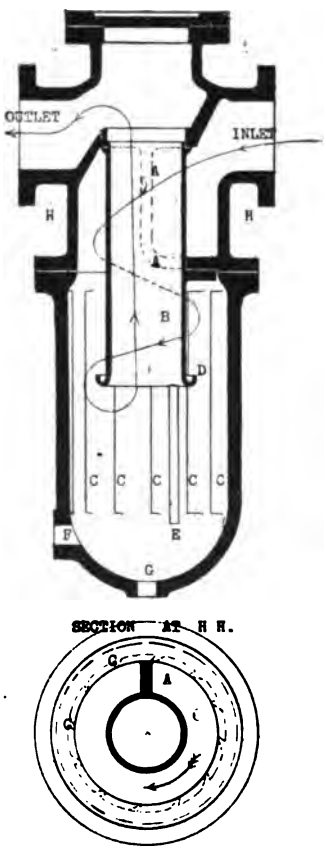


FIG. 214.—Separator or Steam Dryer.

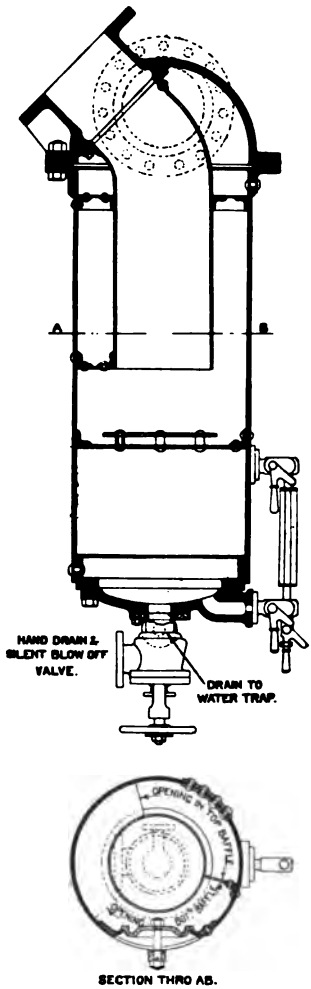


FIG. 215.—Separator and Drain Vessel.

Automatic Separator Blow-Down.—The water separated from the steam during its passage through the separator, and deposited in the bottom of the chamber, is sometimes blown out automatically to the condenser. The automatic apparatus consists of a float

rising and falling with the water-level in the separator, a balance weight, and valve, worked by levers attached to the float. The valve is generally of the balanced or double-beat type. The apparatus is preferably contained within the separator, so that the packing otherwise used for the rod and valve spindles is not required. Such packing frequently upsets the perfect balance of the float and weight, producing variable and unreliable working. A hand blow-through valve is also fitted, and is now usually made of sufficient size for use as a *silent blow-down*. Both the automatic and hand blow-through pipes are connected with the main eduction pipe.

The Belleville separator is also drained automatically. The float is inside the chamber, and the valves outside. It has the advantage of being easily detachable for repair or refit.

Emergency Valve.—In cases of extreme urgency the inter-

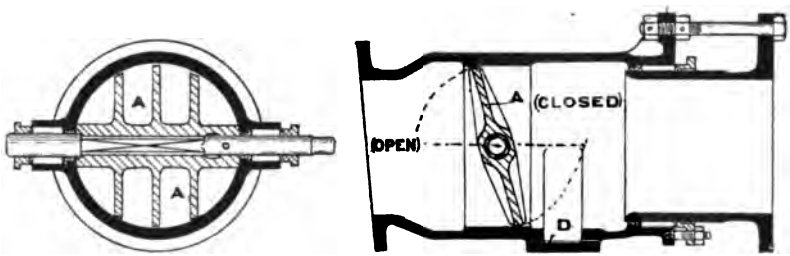


FIG. 216.—Emergency Shut-off Valve.

mediate valve cannot usually be closed with sufficient quickness, and in all recent naval vessels a quick-closing valve is fitted between the intermediate valve and the regulating valve. A type of quick-closing valve is shown in Fig. 216, and consists of a nearly circular disc, *A*, which, when turned into certain position, either leaves an almost uninterrupted opening, or, as shown in the figure, entirely closes the passage-way. The valve can be worked from the engine-room platform and from outside the engine room, and under ordinary conditions is always wide open. The spindle of the valve withstands the total pressure on the valve when closed, and is consequently a substantial fitting; and for the same reason the valve is strengthened by webs, as shown.

Regulating and Manœuvring Valves.—Steam is admitted to the engine either through the regulating valve; or, when only low powers are in use, through a smaller valve, called the manœuvring valve, which is fitted alongside the former, and performs a similar

office. The diameter of the smaller valve is generally about one-fourth to one-fifth that of the main regulating valve, and is suitable for powers up to about one-third full power.

The ordinary valve, with the steam pressing it on to its seating, cannot be opened easily. In modern ships, where the diameter of the valve is from 15 to 20 inches, the total pressure on it is many tons, and consequently a balanced valve is now generally used in conjunction with high pressures and large powers. The balanced valve allows the engines to be handled more easily, and reduces the wear-and-tear on the threads of the screwed valve spindle.

In Fig. 217 a balanced valve is shown with the steam pressing the lower valve on to its seating and the inner (upper) valve away from its seating. The two valves being fitted on the same spindle, the end pressure is nearly balanced. The upper valve (that is, the one nearer the cover *C*) is necessarily slightly larger than the lower valve, to allow the latter to be passed into its place; and there is consequently a slight resultant pressure in the direction of keeping the valve open, when the steam pressure is admitted between the two valves. The spindle is generally packed with asbestos well mixed with black-lead, and in some cases white-metal scrapings are used, and make a working joint. As the spindle is generally below the valve box, it is good practice to raise the stuffing box *B* inside, and so prevent any water standing above the packing, because it conduces to leakage.

When the valves, the spindle, and the valve box are all made of gun metal, or practically the same material, there is no great difficulty in obtaining tightness under ordinary conditions. For large valve boxes cast steel is now frequently used, but the same material is not suitable for the spindle and valves. There is therefore some variation in the length of the spindle and distance of the valve faces from each other at different temperatures, while the distance of the valve seatings is not necessarily exactly similarly affected. This allows leakage, and special methods have been introduced to meet it, one of which is shown in the figure.

The seatings are removable, and jointed to the casting by studs and nuts, and are generally made of gun metal. Stays are fitted as shown, to resist unbalanced pressures on the walls of the valve box.

The *manœuvring valve M* is shown in the figure; the pressure enters to the back of the valve, and although of small dimensions, the total pressure on it when closed is very considerable.

Gearing is fitted to each of the two valves, the regulating and manœuvring valves, to work them from the starting platform. The

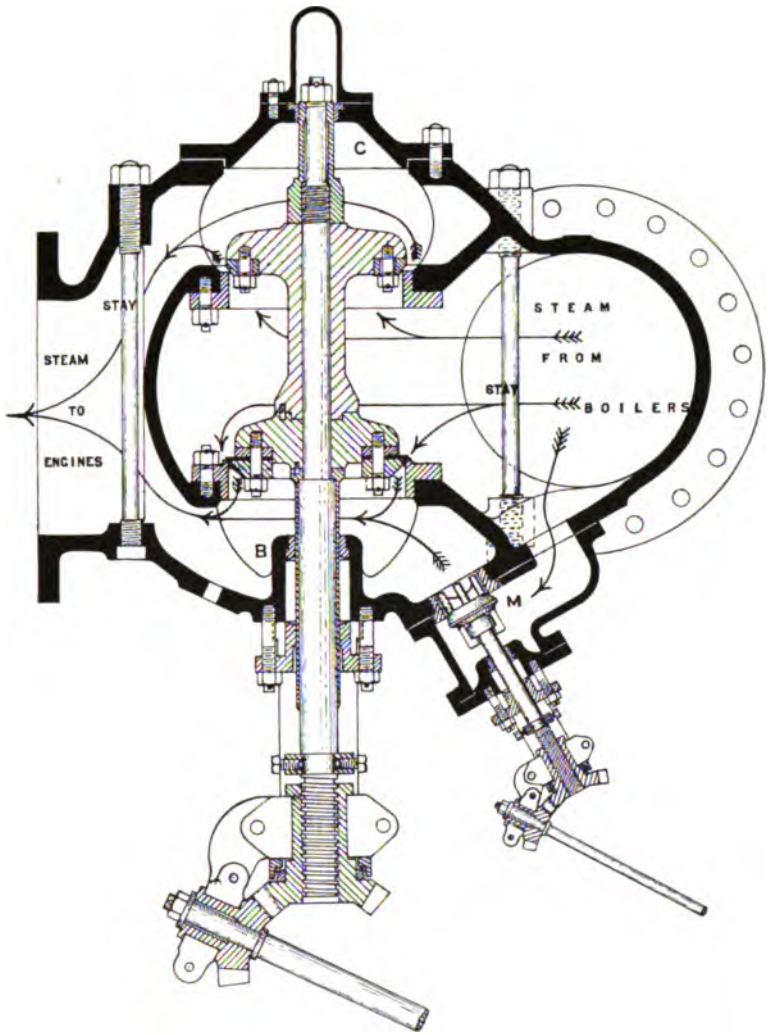


FIG. 217.—Main Steam Regulating and Manœuvring Valves.

larger valve spindles are sometimes fitted with a square threaded screw, but the smaller valves are more frequently fitted with ordinary Whitworth threaded spindles.

A flanged hole is fitted for the safety valve on the H.P. casing, and another for the valve admitting steam to the starting valves and cylinder jackets.

In some of the modern warships the regulating valve is made with the Bevis and Gibson flexible seating, as shown in Figs. 217 and 93. A small pass valve is unnecessary with this arrangement, and constitutes an advantage.

Turbine Manœuvring Valve.—For turbine engines a special

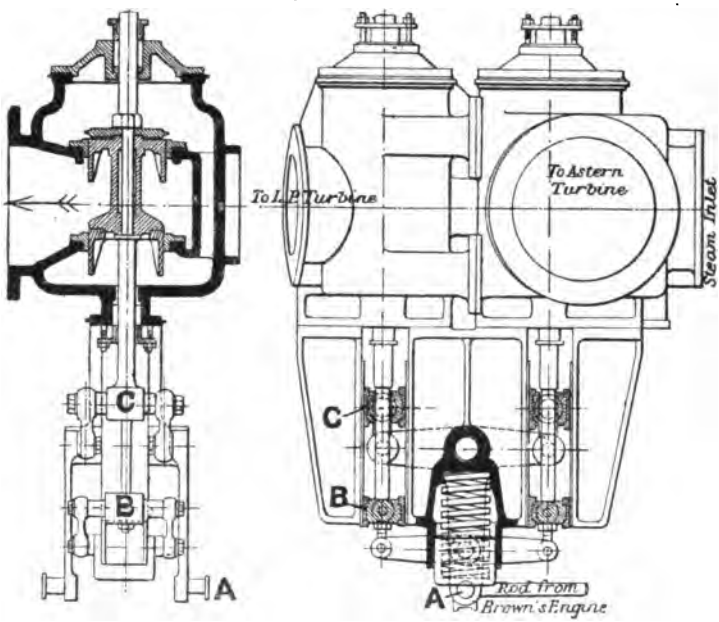


FIG. 218.—Manœuvring Valves (*Mauretania*).

arrangement is required for regulating and reversing the several shafts, and in the *Mauretania* the system shown in Fig. 218 has been adopted for each of the middle shafts in connection with the L.P. ahead and astern turbines. The valves, one for each turbine, are worked by means of wheels on the starting platform. The wheel operates the valve of a Brown's engine (Fig. 186 type), which opens either valve as required without opening the other. When the Brown piston is in mid-stroke both valves are closed; the forward stroke opens the astern turbine valve and the return stroke opens the L.P. ahead turbine valve. The upper of the double-beat valves is of the Bevis-Gibson type and the lower is of the piston

type; and in the mid-position of the actuating gear there is a slight tendency, from the inlet pressure and larger size of the upper valve, for the steam to be shut, and this action is assisted by a spring shown. When the Brown engine pushes the end *A* of the T lever to the left, the astern valve remains closed and steam is admitted to the L.P. ahead turbine. Conversely, if *A* is moved to the right, the L.P. ahead valve remains closed and steam is admitted to the astern turbine. In the *Mauretania* a hand pump is provided at the starting pedestal, which enables the manœuvring valves to be opened and closed. Water pressure from the hand pump thus can operate the Brown engine, in the event of the latter breaking down. The manœuvring valves are used only for entering and leaving harbour, etc., and at other times the steam is shut off the steam inlet to them.

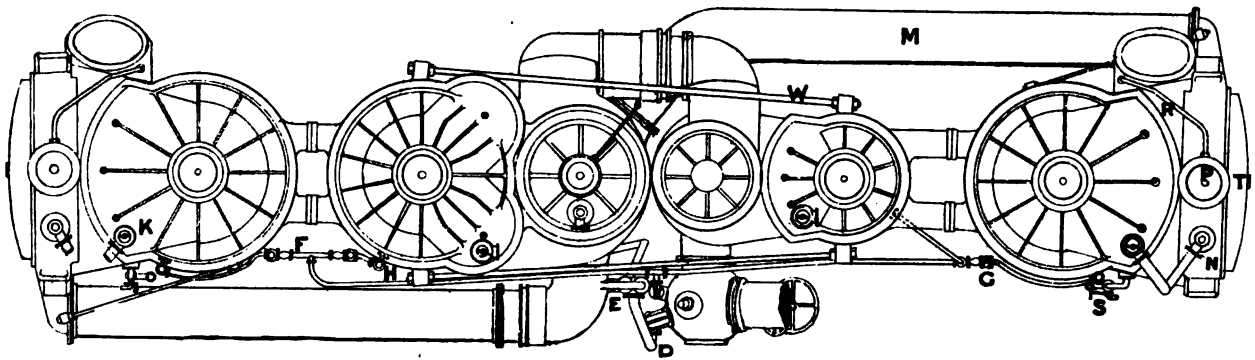
The positions of the main regulating and manœuvring valves are shown in Figs. 219, 220, and the other engine mountings will now be considered in detail.

Starting Valves.—At certain parts of the stroke the admission of steam to the H.P. slide valve will not start the engine, and it is necessary to admit steam to one of the other cylinders to produce motion.

A small D slide valve was originally fitted outside each cylinder for this purpose, and by working a lever, in connection with its slide rod, steam could be admitted to either end of the cylinder as required. Great care was necessary to admit steam to the correct side of the piston for the purpose of starting; and the lever was generally arranged to incline in the same direction, when steam was admitted, as the connecting rod for *ahead* motion, and it would therefore be crossed with the connecting rod for going *astern*. With very fast-running engines the operation is extremely difficult; and for all recent types of stage-expansion engines pass valves are fitted, and in some cases in addition to the auxiliary starting valves.

The **bye-pass** starting valve is simply an ordinary stop valve, which can be worked by a lever from the starting platform. One is fitted to each M.P. and one for the L.P. cylinders. Steam can be admitted to the respective receivers, which is then diverted by the slide valve to the correct side of the piston for the required direction of rotation. There is little probability of locking the engines by using the pass valve, although the admission of steam to

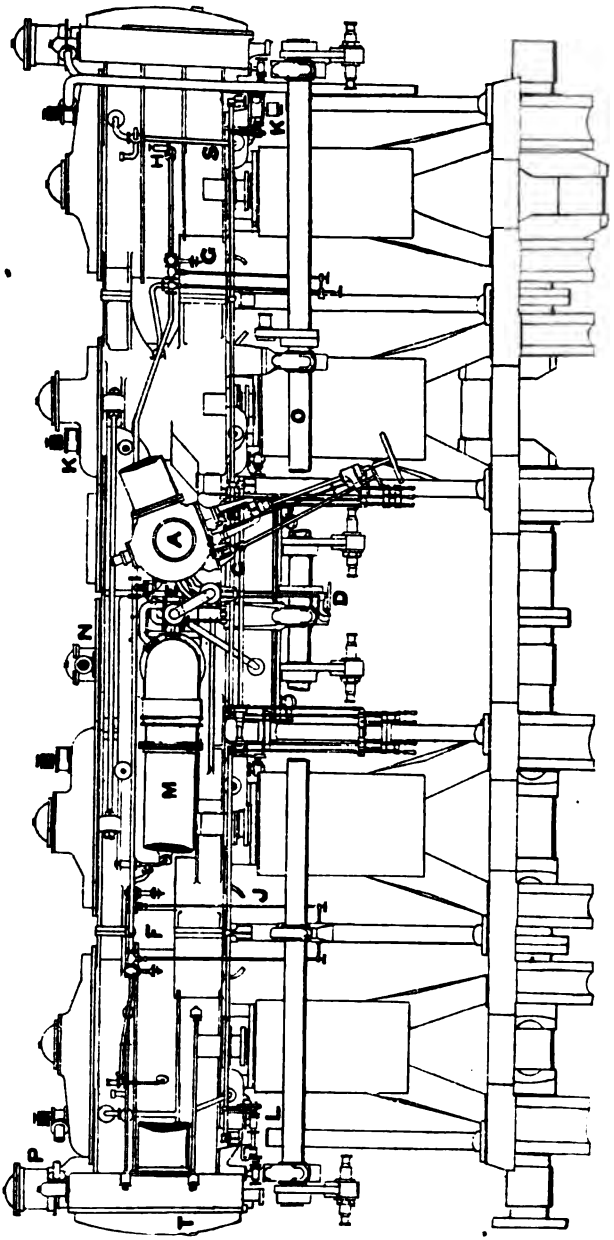
POSITION OF ENGINE MOUNTINGS—PLAN.



ENGINE MOUNTINGS

FIG. 219.

A, Main steam regulating valve.
B, Pass valve for *A*.
C, Manœuvring valve.
D, Steam to starting valves.
E, Starting valves and pipes.
F, Steam supply to jackets.
G, Reducing valves for jackets (one on M.P. and each L.P. cylinder).
H, Safety valves for jackets (one on M.P. and each L.P. cylinder).
J, Drain pipes to jackets (one below each cylinder).
K, Cylinder relief valves (one above and below each cylinder).
L, Drain cocks (one below each slide valve and cylinder).
M, Receiver pipes, with expansion joints.
N, Safety valves, in connection with receivers (one above M.P. and each L.P. slide chest).
O, Weigh shaft for reversing gear.
P, Balance cylinders (one above M.P. and each L.P. slide chest).
R, Pipes connecting balance cylinders with eduction and receiver pipes.
S, Indicator pipes and cocks (one for each cylinder).
T, Steam gauge connections for equilibrium rings of L.P. slide valves.
W, Ties or stays between cylinders.



POSITION OF ENGINE MOUNTINGS—ELEVATION.

FIG. 220.

the receiver may retard the motion of the piston fitted in the next stage *above* it. The piston in the stage below, to which steam is admitted through the slide valve, is larger, and should generally overcome the possible retardation.

In a few particular instances the bye-pass valve is not always suitable. Thus, in a two-cylinder two-stage expansion engine, with the cranks set at right angles, if the H.P. crank is on the dead centre, the L.P. crank is at about half-stroke, and steam will not be admitted through the slide valve by the bye-pass unless the cut-off is later than half-stroke. This is exceptional, but the modern four-cylinder engine is sometimes similarly constructed.

Cylinder Escape Valve.—When, from any cause, an extra pressure or an accumulation of water finds its way into the cylinder, it is allowed to escape through the relief valves fitted on the cylinder above the top end, and below the bottom end, of the piston stroke. In vertical engines the valves are generally placed at top and bottom in places where they do not come in contact with the working liner, and the valve box is fitted with a spigot entering the hole in the cylinder. A *safety valve* is also fitted above the slide chest, to prevent the accumulation of too great pressure. This is particularly necessary for the M.P. and L.P. cylinders, because they are designed to carry a reduced pressure, not the full working pressure, as in the case of the H.P. cylinder.

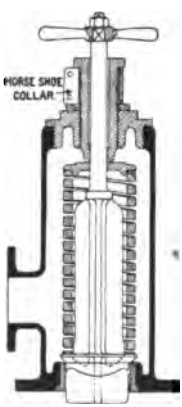


FIG. 221.—Relief Valve.

The escape (or relief) and safety valves are of the same type as a boiler safety valve, and their construction is similar. There is usually a valve in solid connection with its spindle, as shown in Fig. 221, which is fitted with the usual T handle. A spring, fitted to the valve box, and of square section, loads the valve to the required pressure, which is generally a few pounds above the highest working pressure used in the cylinder. The safety valve emits to the upper part of the engine room; but the relief valves, being connected by a common pipe led down to a safe point below the working platform, generally open into the crank pit.

An accumulation of water in the cylinder is dangerous, as it is practically incompressible; and if the escape is cut off through

the slide valve, the relief valve allows it to pass harmlessly into the bilge.

Indicator Pipes.—A large pipe, about 2 inches in diameter for a battleship engine, is fitted to each cylinder, and is connected with the spaces above and below the piston stroke. At the upper, or any convenient part, a two-way asbestos-packed cock is fitted in the pipe, and has a projection fitted in connection with it for use with the indicator. By means of the cock, connection can be opened with either end of the cylinder as required for the indicator to take readings. A system of levers is fitted from the piston-rod end for working the indicator in unison with it, and one of these levers is generally used for working the engine revolution counter.

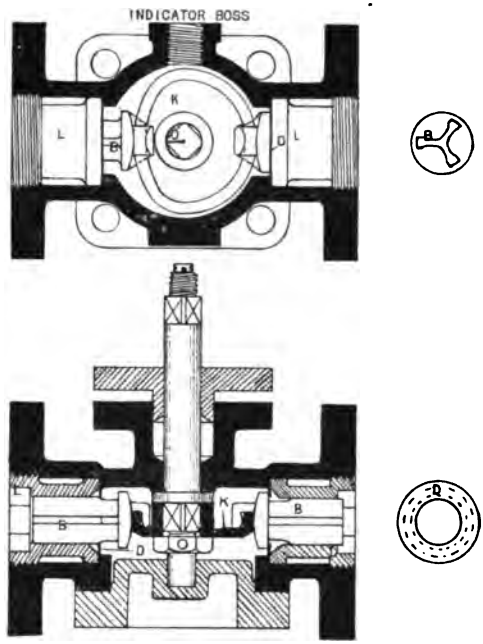


FIG. 222.—Bevis-Gibson Indicator Pipe Valve.

In Fig. 222 a type of valve for use with the indicator is shown. The seating is flexible, and made of thin sheet-metal, in the shape of a ring, as at *D*. The valve is of ordinary construction, with three webs, as at *B*, with a narrow flat seating; above the valve a lug is made, catching over the lip of the cam *K*, which moves the valve through its means. The cam is formed with a spindle, which is worked from outside the valve box, as in the case of an ordinary asbestos-packed cock, and when one valve is closed the other may be either closed or open. A mark on top of the spindle shows the direction in which the opening is pointed. The seating is fixed in position by a valve case, *L*, which screws into the valve box. The valves can be inserted or removed either by removing the bottom cover or by taking down the valve box and removing the seating. The pressure keeps the flexible seating in tight contact with the

valve, and thus obtains a steam-tight joint. A screwed hole is fitted on top of the valve box for inserting the indicator.

A similar arrangement is used for other purposes, as a multiple way valve, and it is then made with a flexible valve as well as seating. The flexible valves are Bevis and Gibson's patent.

Jacket Steam.—The steam for the cylinder jackets is generally taken from the steam side of the main regulating valve, and a shut-off valve is fitted at this end of the pipe to control the supply to all the jackets. A branch is led from this supply pipe to each cylinder jacket, and each of the supplies to the M.P. and L.P. jacket is controlled by a separate stop valve, a reducing valve (to keep the pressure below that for which the cylinder is fitted and made), and a safety valve (which gives warning to the engine room, should the reducing valve fail to act).

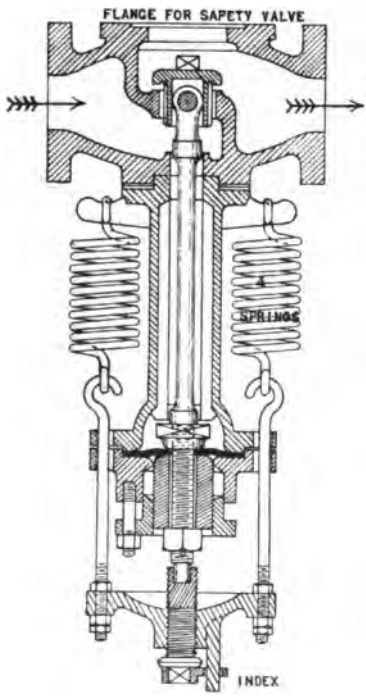


FIG. 223.—Reducing Valve.

(The steam for the starting valves is also frequently led from the same pipe that supplies the jackets.)

A separate supply is fitted for the H.P. cylinder jacket of warship engines, and is led from the slide chest independently to the jacket space. The pressure and temperature in this way cannot exceed that due to the maximum pressure in the cylinder, which, if excessive, frequently causes undue friction of the working parts. The control valve for this supply must be non-return, or the pressure from the other supply to the jacket might escape into the slide chest.

Reducing Valve.—A type of reducing valve suited for the jacket steam is shown in Fig. 223. The springs are in tension, and tend to keep the valve open at all times. When the pressure on the reduced-pressure side of the valve exceeds that for which the springs are set, the valve closes, and prevents the pressure rising. The inlet pressure tends to keep the valve open,

and this pressure is balanced by the pressure in an opposite direction on an india-rubber ring, shown in black section, and consequently the opening or closing of the valve depends on the springs and the pressure on the reduced side of the valve. The tension on the springs can be regulated to a moderate extent by a screw fitting over the lower end of the valve spindle, the movement and pressure being shown by an index.

Steam Drain System.—The drainage of the steam and exhaust systems is much simplified by sloping the main run of pipes gradually, without pockets or dips, from the source of supply to some convenient depository, such as the separator and condenser, and in the more recent warships this is usually done. Separators are fitted to the main steam pipes, as already described, and to the auxiliary steam and exhaust pipes at their lowest parts in each engine room. In addition, drains are fitted to the valve boxes connected with these pipes, and led, through drain traps (see page 416), into the steam drain pipe common to the machinery and boiler spaces.

The water in this steam drain pipe gravitates into a tank, fitted in the engine room or boiler room as convenient, whence it is removed by a pump, which delivers into the grease extractor or feed tank as required. The tank is closed and fitted with a relief valve, manhole, and cover and drain cock, and a float and index, to show the quantity of water in it. The drain water runs into branches connected with this drain pipe at various parts of its run.

The safety-valve boxes are drained into open-topped drip tanks, which overflow into the drain pipe, and so to the drain tank. The steam cylinder and slide-valve chests of the auxiliary engines are drained into the bilge, and not into the drain system. By an arrangement of one or two of the branches to the auxiliary engines from the steam pipes at various parts of the ship, a fall can be made, which serves as a drain for the general run of pipe; and the drain pipe from this branch, through a drain trap and into the tank, relieves the pipe of all accumulations of water.

Jacket Drains.—In some of the older warships each cylinder-jacket drain pipe is connected with a separate vertical cylindrical gun-metal tank, generally fixed to one of the columns supporting the cylinder. A water gauge is fitted to the tank, and any water accumulating can be blown out as required into the main eduction pipe. If practicable, however, this drain water should be used to

warm the feed water, and thus promote some economy. The heating of the feed water to the moderate temperature which is obtained by the drain water does not reduce the efficacy of the pumps if the air and vapour expelled by the effect of heat is allowed to escape. The drain should be fitted with a nozzle inside the tank, to effect the heating quietly, without shock, and in a definite direction, for circulating the water inside the tank.

An unnecessary loss of heat can be prevented by allowing the condensed water to accumulate in the cylindrical tank, and keeping this at a regular working level by adjusting the drain cock. As only water is then blown out, very little heat is wasted. In the mercantile marine, and generally on shore, the jacket drains are led through steam drain traps, which automatically serve the above purpose.

Cylinder Drains.—When raising steam, or the engines are standing under steam, there is some condensation in the cylinders, and the water is not swept out by the exhaust, which generally happens when the engines are moving. The drains from each of the H.P. and M.P. cylinders and slide-valve chests are fitted with a cock close to and below the cylinder, which can be worked from the starting platform. These cocks are in connection with separate pipes, leading to switch cocks, which can be connected with either the main condenser or the bilge. The leads into the condenser are fitted with shut-off valves at the discharge end, and the leads into the bilge are cut off at a convenient height, so as to be visible from the starting platform.

The same arrangement is fitted for the L.P. cylinder and slide-valve chest drains, but the discharge is conducted into the steam side of the condensers and the bilge. The lead to the condenser is inclined downwards throughout its entire length, and a shut-off valve is fitted on the condenser end. The lead to the bilge is fitted with a non-return valve, near the switch cock, and serves to test the presence of water, or otherwise, in the cylinder.

Steam Drain Traps.—Two principal types of drain traps are in common use both for marine and land purposes. Generally speaking, the float type is used for moderate pressures and the expansion type for the highest pressures; but the Lancaster trap, which is of the *float type*, is fitted in the *Lusitania* and *Mauretania* for a working pressure of 195 lb. per square inch. This apparatus is of very simple description. A tank or trap is formed below and in

connection with the part to be drained. Inside the tank is a float which automatically controls the drain valve, and thus allows the water to escape into the drain pipe system when a rise of level of water raises the float sufficiently.

An *expansion drain trap* works on the principle of different rates of expansion of two component parts. Academically, the temperatures of steam and water in the same vessel and under the same pressure are the same; an expansion drain trap would not work if this were always absolutely true. Actually, any water contained in the apparatus cools more rapidly, due to its greater conductivity, than the steam from which it is condensed, and therefore this cooling produces a contraction of a tube containing water, and opens a valve.

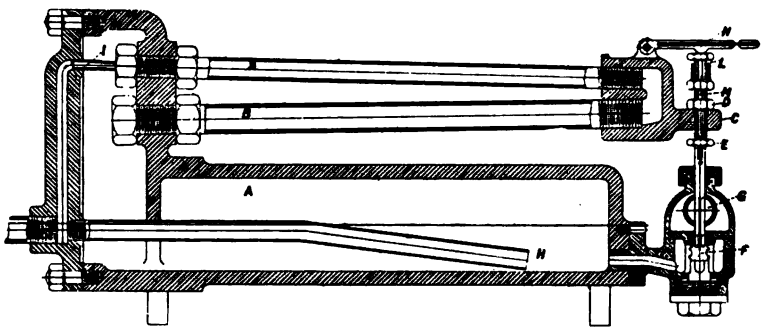


FIG. 224.—"Patros" Steam Trap.

In Fig. 224 a section through a "Patros" steam trap is shown. *A* is a reservoir on the drain trap for storing the collected water which accumulates until it fills the tubes *B*. The lower tube is of metal and the upper one of iron. Cooling of the tubes produces a difference of contraction, and draws down the end *C* until the space intervening between the nuts *D* and *E* on the spindle is traversed; further contraction pushes the valve spindle downwards and opens the valve, which allows the water to escape from *A* through the pipe *G* until the water-level falls below the inlet pipe *H*. Steam then passes through the bye-pass *I* into the upper tube *B*, and circulates through the lower tube into *A* until all water is discharged. Both tubes expanding raise the end *C*, and the valve is closed by the steam pressure. Prompt closing is ensured by a spring, *L*. A lever, *N*, and pin, *M*, are fitted for testing the trap by hand.

Another arrangement is shown (Holden and Brooke's). In Fig. 224A the drain pipe is connected with the interior of a long brass tube, *B*, through the inlet *A*. Near the inlet a yoke, *C*, is fitted around the tube, against which two rods, *E* and *F*, are pressed, and the pressure on the yoke is supported by studs, *S*, fitted behind it and attached to the tube. The rod *F* is butted at its left-hand end against the automatic drain valve box *G*, and any contraction or expansion of the tube tends to tilt the yoke. The left-hand end of the rod *E* butts against a lever, *L*, which presses the valve open, or, when the tube is expanded in length by heating, is kept pressed

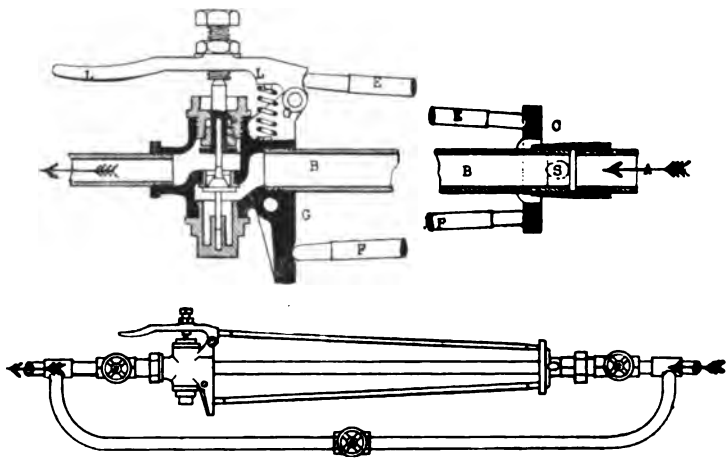


FIG. 224A.—Brooke's Steam Trap.

against the yoke *C* by a spring under the valve lever *L*. When the tube is cold it contracts, the yoke is tilted by the rod *F*, and the rod *E*, which also tends to tilt the yoke in the opposite direction, is constrained to overcome the spring and open the valve, and thus allows the water contained in the tube to escape, until the entrance of steam warms the tube sufficiently to close the valve. A bye-pass pipe is fitted between the inlet *A* and the discharge, as shown in the lower sketch, so that the trap can be examined or repaired at any time; and a test cock is sometimes fitted on the discharge side of the automatic valve. The lever which works the automatic valve is formed so that it can be worked by hand when required. The great advantage of expansion steam drain traps is that the discharge is always wide open when the pipes are cold, and no

accident then occurs from the drains being forgotten and left unopened.

The arrangements above described are suitable for high pressures and great range of temperature without adjustment. There are certain distinctive features:—

1. The discharge is intermittent, and thus free from dribbling through a very slight opening of the valve, which tends to destroy the seatings.

2. Before the valve opens, the contraction of the tube must be sufficient to overcome the spring and the unbalanced pressure, exerted by the steam and water on the valve, tending to keep it closed. Directly any opening begins, the pressure, being relieved, allows the valve to open a *definite* distance, not slightly or gradually, and thus allows a definite discharge.

3. When the expansion of the tube allows the valve to close, it does so in the same definite manner, and is kept closed by the pressure behind it, not by the expansion.

4. The certainty of definite action allows the discharge to be led directly into an open-ended pipe, such as is occasionally fitted in connection with the steam drain system and tank.

5. Brooke's trap acts equally well in any position, either vertically or horizontally, and can be conveniently fitted to drain steam pipes, cylinders, and jackets.

6. The valve-spindle packing (Brooke's) is protected from the direct impact and scouring of the outflowing water by an inner flange (shown in the figure) round the spindle.

7. The interval and amount of discharge can be regulated by adjusting the set screw and lock nut fitted to the lever.

8. The "Patros" trap allows a range of pressure, without waste from blowing steam directly through, of over 60 lb., and the temperature of discharge is near 212° F.

Another form of steam drain trap is made by Holden and Brooke, known as the Sirius type, and is suitable for pressures up to 120 lb. per square inch. Its action depends on the movement produced by expansion and contraction of a tube of the Bourdon-gauge type, and when seen can be at once understood.

CHAPTER XIX

THE FEED WATER AND PUMPS

Sources of Supply.—The water condensed in the condenser is pumped out of it by the air pump, which delivers it into either the hot-well tank, from which it is pumped by the hot-well pump into the feed filter, or into the filter direct; from the filter it flows into the feed tank. From the feed tank the boiler feed pumps take their supply and return the water to the boiler. The feed tanks are also supplied from the auxiliary air pumps and from the drain tank by its pump, but usually both these supplies are passed through the filter on their way to the feed tank.

The blow-down, which is generally automatic, and fitted on the steam separator in the steam pipe just previous to its connection with the engine, is also a source of supply to the condenser as a water-saving appliance. Any surplus steam can be passed through it, as a silent blow-off, and the water, instead of being lost up the waste-steam pipe, is saved by passing it through the condenser.

The water-saving appliances are thus—

1. Main and auxiliary condensers.
2. Silent blow-off, or automatic drain to main steam pipe.
3. Steam drain system; the water is collected in a drain tank from steam traps fitted at various parts, and pumped through the filter to the feed tank.

The whole of the water which leaves the boiler as steam cannot all be saved; there must be some loss from leakage, and this is made up from the reserve feed tanks or from the evaporators.

Necessity of Fresh-water Feed.—Sea-water contains about $\frac{1}{32}$ part by weight of solid matter, in which common salt predominates but is not generally deposited on the heating surfaces of the boiler, and is easily removed by blowing down.

Sea-water contains about, in lb. per ton (2240 lb.),—

58	of common salt (chloride of sodium).
5	„ sulphate of magnesia.
8	„ chloride of magnesia.
$\frac{1}{4}$	„ carbonate of lime and magnesia.
$3\frac{1}{2}$	„ sulphate of lime.
$1\frac{1}{4}$	„ organic matter.
Total 76 lb.	

Of these, the chloride of magnesia is the most dangerous. At a temperature of 212° it decomposes into magnesia and hydrochloric acid. The latter is an active corrosive agent; but the corrosive effect is neutralised by adding alkalis, soda, or lime to the feed water, which should always be alkaline.

Sulphate of lime (calcic sulphate) is the most troublesome deposit, with the exception of oil. At a temperature of about 280°, corresponding to a pressure of about 40 lb., sea-water is incapable of holding sulphate of lime in solution.

Carbonate of lime forms, with the sulphate of lime, about 85 per cent of the scale usually deposited in the boiler. The carbonate, although it is not freely deposited until a temperature of 350° is reached, corresponding to a pressure of 120 lb., is, however, deposited at a lower density than the sulphate, and is generally first deposited on the heating surfaces. It is followed by sulphate of lime, salts of magnesia, silica, and common salt, in the order named; but the two last are very seldom found, unless the density has been very high, and above that usually maintained, even when the feed water is nearly of the consistency of sea-water.

About 1 lb. per ton weight of sea-water, density 10, fed into the boilers is left as scale deposited on the heating surfaces, when the boilers are blown down at regular intervals according to the regular practice.

The rate of conduction of heat through a clean lime scale is only $2\frac{1}{2}$ per cent of that of steel, such as is used in a furnace. By practical experiment it has been shown that with $\frac{1}{16}$-inch of scale 15 per cent more fuel is required, and with $\frac{1}{4}$-inch of scale about 60 per cent. With a perfectly clean furnace of ordinary thickness, the temperature of the water surface of the plate may be only 15° in excess of the steam generated; but with a scale of $\frac{1}{2}$-inch

thickness the temperature of the surface must be raised to nearly double that of the steam generated. With an oil scale, the temperatures are exceedingly high in the metal of the furnace compared with that of the steam generated. It is an oil scale which is the greatest source of danger in boilers, and for continuous working a grease filter is necessary.

At temperatures above 600°, steel tends to become carbonised into cast iron with a consequent reduction of tensile strength.

It is for these reasons (the prevention of scale forming and consequent overheating of the boiler plating, and for promotion of economy) that sea-water is excluded as much as possible from high-pressure boilers.

Quantity of Water pumped.—The weight of steam used must be returned to the boiler as feed water to maintain the same safe working level. This amount may be taken as 12 to 20 lb. per I.H.P. per hour. For example, take the I.H.P. as 11,200, and the weight used per I.H.P. per hour as 15 lb. Then in every 24 hours the total feed water pumped is—

$$11{,}200 \times 15 \times 24 \div 2240 = 1800 \text{ tons};$$

or, quantity of water pumped is 160 tons per 1000 I.H.P. per 24 hours.

Feed-water Loss.—Under good conditions of working, and with well-appointed saving appliances, the loss should not exceed about $3\frac{1}{2}$ tons per 1000 I.H.P. per 24 hours, so that the actual loss of water pumped is only about $2\frac{1}{4}$ per cent. Unfortunately, the loss is frequently greater, and, with high pressures, the tendency to leakage is also greater. Ships vary considerably in the arrangement of machinery and steam pipes, and the number of auxiliary engines is very great, particularly in warships. This method of computing the loss is very unreliable, but it is the nearest yet suggested, and with improved simplicity of arrangement the error becomes less.

For naval vessels, the following amounts of loss must not be exceeded on trial :—

5	tons	per 1000	I.H.P.	per 24	hours,	at 20 %	of the full power.
$4\frac{1}{2}$	tons	„	„	„		at 70 %	„ „
4	tons	„	„	„		at 100 %	„ „

During the contractor's trials of ocean-going t.b. destroyers the loss was generally under *one* ton.

The evaporators are generally capable of producing, *when clean*, about 10 tons per 1000 I.H.P. per 24 hours.

Hot-Well and Pump.—In consequence of the increased work thrown on the air pumps when they were also required to deliver the condensed water through the filter into the feed tank, it was considered advisable to fit a pump for this special purpose, and so relieve the air pump of the unnecessary stress. The air pump therefore delivered its contents into the hot-well, which was fitted in close proximity, and the hot-well pump then forced the water up to the filter (with a pressure filter it also forces it through the filtering medium), from which it flowed into the feed tank.

The hot-well, if fitted, is of small capacity, and an automatic arrangement is generally fitted to the pump, so that it is governed by the height of water in the tank. The steam-cock of the engine driving the pump is attached to a system of levers, which are operated by a float inside the tank; any increase of water in the tank opens more steam to the engine, and any decrease has an opposite effect. An air outlet pipe and an overflow pipe are both fitted to the tank, so that, in the event of its becoming too full, the water can escape to the feed tank or reserve tank. Generally one hot-well and pump are fitted in each engine room, when arranged for their use. The adoption of a gravity filter, a better type of air vessel on the air pump, and finally the adoption of an independent air pump, have each and all contributed to the abolition of the hot-well and pump.

Feed-water Filter or Grease Extractor.—The filter takes its supply from three sources:—

1. Main air pump, or hot-well pump, if fitted.
2. Auxiliary air pump, if fitted; with the independent air pump it is unnecessary.
3. Drain-tank pump.

There are several varieties of filters, but the one now usually adopted is of the gravity type. Under favourable conditions there is no necessity for a hot-well and pump, as the filter itself takes the place of the former, which simply acts as a quiet place in which the air can be evolved from the water and escape to the outer air or atmosphere.

A *gravity filter* is generally placed as near as convenient to the

air pump and at a fair height above the feed tank, so as to give a steady gravitation of the water into the latter and an uninterrupted and non-devious flow through the air-pump discharge.

Inside the Rankine filter (Fig. 225) there are several cells, on frames, in each of which three separate pieces of filtering material, generally of coarse towelling, are fitted. The cells are of circular shape. A water-tight cover is fitted on the end of the filter, and is fastened in its place by hinged screws and butterfly nuts. After removing the cover, the cartridges, or any separate stratum, can be removed, and new filtering material can be fitted. A spare set of

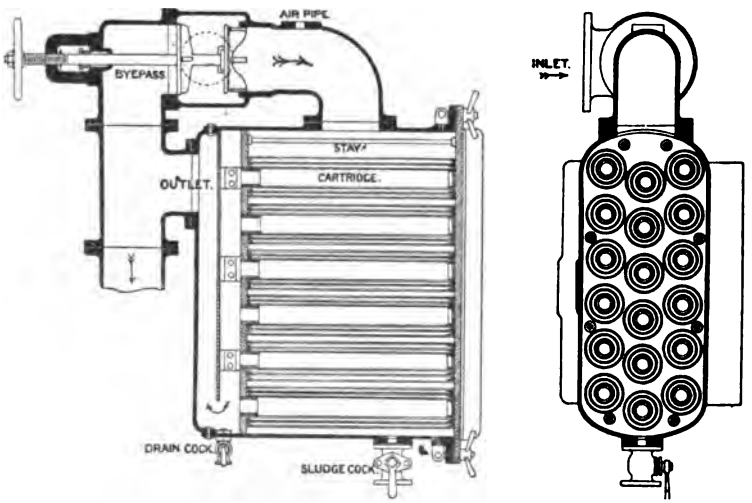


FIG. 225.—Rankine Feed-water Filter or Grease Extractor.

cartridges is supplied, and can be put in at once; this operation should only take a few minutes if everything is previously prepared. An air pipe is led away from the highest available part of the filter to a height of about 4 feet. About $1\frac{1}{2}$ to 2 square inches of total surface of the three strata through which the water passes are allowed per I.H.P. A bye-pass valve is fitted for use when opening the filter.

As shown in Fig. 226, the main inflow pipe to the filter is fitted with a shut-off valve, and when this is shut the water is pumped to a higher level and passes through a pipe connected directly with the outflow pipe to the feed tank. This pipe prevents any great influx of water overloading the air pump or filter.

Any filter which can be used as a *pressure filter* can also be used as a gravity filter. The pressure filter is situated between the hot-well pump and the feed tank, and the water is *forced*, in contra-distinction to flowing by gravity, through the filter. A bye-pass pipe and relief valve are fitted, so that the water can pass when the filter is not in operation. A pressure gauge is fitted to the inlet side, and also a scum chamber, by which any floating oil or sludge can be drawn off. When the cloths become foul it is shown by the increased pressure on the gauge, which usually shows about 5 lb. pressure when the cloths are clean. Before cleaning, the inlet and outlet valves are closed, a little steam at a low pressure is admitted, and the grease is blown away through the sludge outlet. The cleaning should not take more than two or three minutes, and afterwards the pressure, as shown by the gauge, should be at the usual point. The pressure filter is not so efficient a grease extractor as a gravity filter. The power required to force the water through it is unnecessarily great, and if the extra work is done by the air pump the type is objectionable, because the air pump is primarily a suction pump, or air extractor, and not a force pump. There are some cases where a pressure filter is more useful, because it can be rather smaller, and fitted where a larger filter would be impossible.

With boilers working at very high pressures a good filter or grease extractor is a necessity, and the extra cleanliness of the boiler produces economy of fuel and repairs.

Feed Tank and Fittings.—The feed water gravitates from the filter, or it is pumped, in the case of a pressure filter, into the feed tank. One feed tank is fitted to each main engine, and they are connected with each other through a large pipe with a shut-off valve at each tank. The total capacity is about $1\frac{1}{4}$ to $1\frac{1}{2}$ tons per 1000 I.H.P., which is equal to about 10 minutes' supply for the feed pumps and boilers when the engines are working at their full power.

The general arrangement of the feed tank and its connections is shown in Fig. 226. They are described below, with their uses :—

(*a*) Gauge glass, extending from nearly the top to the bottom of the tank, to indicate the quantity of water in it.

(*b*) Index plate, fitted near the gauge, to show the quantity in *tons* of water in the tank.

(*c*) Connection with tank in other engine room, with a shut-off

valve next each tank. In some earlier vessels only one valve (a sluice) is fitted.

(*d*) Inflow pipe from the grease extractor, or filter. Means are sometimes fitted by which the water can be diverted into either tank.

(*e*) Filling pipe from alongside, led into the top of the tank, and fitted at some point above its upper part with a steam injector

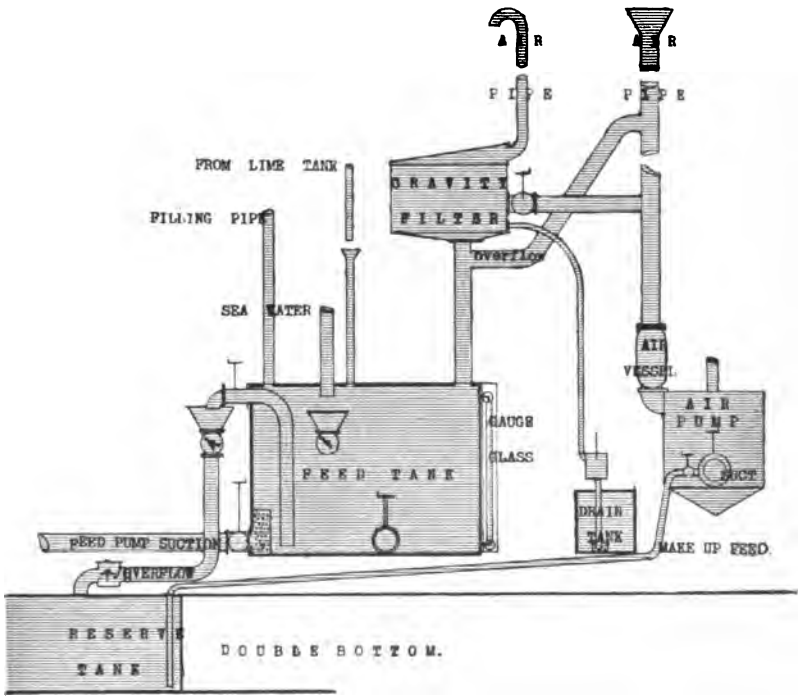


FIG. 226.—Feed Tank and Connections.

and pipe for drawing the water up from a tank or vessel alongside, and discharging it into this pipe through a bell-mouthed funnel, and so to the tank.

(*f*) Supply from the sea, led into a funnel-shaped vessel on the side of the tank, and fitted with a shut-off valve. The water is obtained either from the fire main discharge by connecting on a hose, or by a pipe fixed in a convenient position for the purpose and which can be diverted from dripping into the funnel when it is not required. The top of the funnel is kept below the overflow

from the feed to the reserve tank, so that sea-water cannot enter the latter.

(*g*) Supply from the lime tank, led into the top of the tank, preferably through a funnel, so that the passing water can be seen, as with the alongside and sea-water supplies. The lime is precipitated in a small tank in the upper part of the engine room, and the clean lime water gravitates from the upper part of this tank into the feed tank, leaving behind as much as possible of the sediment.

(*h*) Overflow to reserve tanks, fitted with an internal pipe leading to the bottom of the feed tank so as to allow as far as possible only clean water to pass through the overflow. A shut-off valve is fitted at the outlet, which is led into a funnel and pipe leading to the reserve tanks; a shut-off valve is fitted just below the funnel, and a non-return valve is fitted next the reserve tank. Arrangements are made so that any one of the reserve tanks can be filled by opening or closing the proper valves.

(*i*) Feed-pump suction pipe, fitted to connect the bottom of the tank with the feed pump, with a necessary cross connection for pumping from either tank with any pump. A shut-off valve is fitted next the tank, and a strainer inside the tank over the pipe orifice. The position of the feed tank should be such that there is a free flow through the pipe into the pumps; the pump suction boxes are therefore placed as low as conveniently possible, and the feed tank about 3 feet above the suction boxes of the pumps.

The sides, ends, and top and bottom of the tank are flat surfaces, and it is generally necessary to fit stays horizontally from side to side, but only occasionally from end to end and from top to bottom. A manhole door, fitted internally, and similar to those used for boilers, is also fitted. For protecting the plating from galvanic action, slabs of zinc are connected electrically with the stays or other convenient parts. A drain valve is fitted for use when required to totally empty the tank. If possible, it should be in sight from the starting platform, so that any leakage may be seen.

Reserve Feed Tanks.—In nearly all cases where the ship is built with a double bottom, some of the spaces thus created are utilised as tanks for the storage of fresh water for use as boiler feed. The spaces used are generally situated about the mid-length of the vessel; in cargo steamers the forward and after ballast tanks are also used, as convenient.

The arrangements for filling, for recent warships, are as already described, *i.e.* through the feed tanks from alongside; the gained water from the evaporators passes through the condenser, feed tank, and overflow, with the other condensed water. Open-ended air outlet pipes are led from each tank at about their highest points. The auxiliary feed pumps can draw water from any one tank by opening suitable valves, and also from the feed tanks; but the main feed pumps usually pump from the feed tanks only. A safety valve, loaded to about 50 lb. per square inch, is fitted to each pump suction to prevent overloading from any back pressure, and so bursting the pipe.

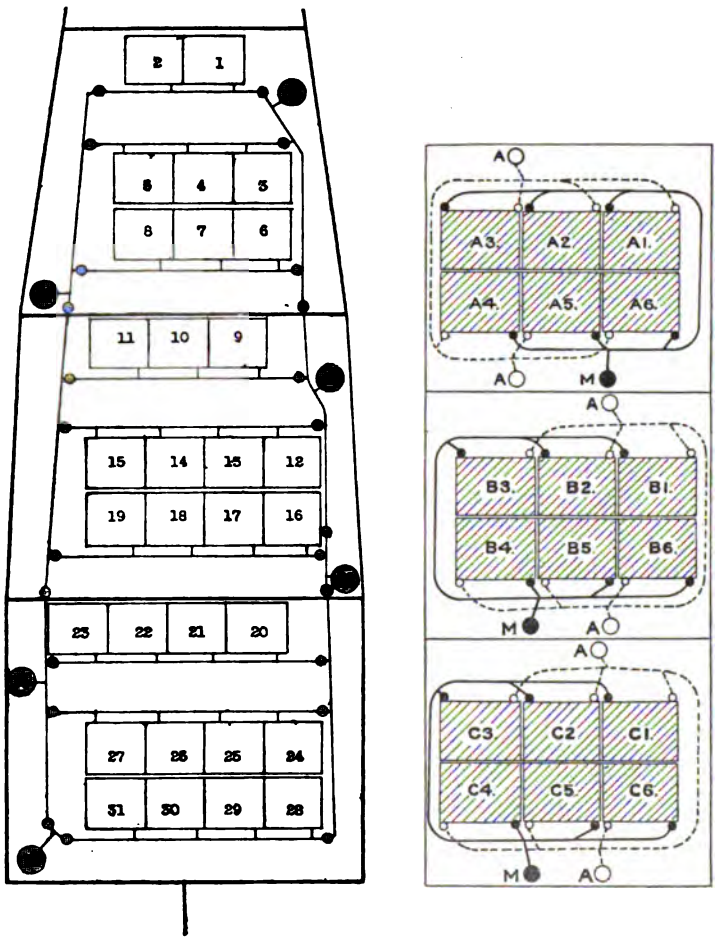
Zinc protectors are suspended in each tank, as in the feed tanks, fitted in electrical contact with the ship plating. The reserve and feed tanks are not painted internally, but kept clean and free from accumulations of oxide scale; and after cleaning, they are rubbed down with mineral oil and dried off as much as possible with a wad. Care must be taken to prevent any loose waste or oakum remaining to choke the feed suctions.

Feed Delivery Pipes.—The general arrangement of the feed delivery pipes and pumps is shown in Fig. 227 for an installation of thirty-one boilers of the Belleville type.

There are, in this case, three boiler rooms, and the arrangement is similar in each. In each boiler room there are two feed pumps, which draw their supply from either the feed tanks or reserve tanks as required. Each pump delivers into a common pipe, which makes the circuit of the rooms. The letter *M* denotes the position of the main, and the letter *A* that of the auxiliary feed pump. A shut-off valve is fitted in the pipe connecting each pump with the delivery pipe. Valves are also fitted at each end of each cross-connection pipe, and also at each part where the pipe passes from one room into the next, at the bulkhead. It can be seen that any one pump can be made to supply any one boiler, or that any group of boilers can be isolated from the remainder. The Belleville boiler is supplied through the automatic feed regulator, one only to each boiler, and a shut-off valve is fitted below the regulator, so that each boiler can be isolated from the others. With other types of water-tube boilers, two pipes are led from the supply into separate feed valves on the boiler, one of which is generally independent of the automatic apparatus.

Fig. 228 shows a more recent arrangement for eighteen water-

tube boilers. There are two auxiliary, *A*, and one main, *M*, feed pump in each boiler room. The main and auxiliary systems are entirely independent of each other, and each boiler room is self-contained.



FIGS. 227 and 228.—Arrangement of Feed Pumps and Delivery Pipes.

Feed Pumps.—The feed pumps are a necessary adjunct of the main engines, as it is by their means that the water is returned to the boilers after condensation from the steam supplied to the engine; it thus completes the cycle of operations.

1. The earliest types of marine engines were generally fitted

with a lever attached to, and worked by, the end of the piston rod; this lever worked the main feed pump, and in consequence the pump only worked at the same time as the engine. For supplying the boilers when the main engines were not at work, a donkey pump was generally fitted in the stokehold, and worked by an independent engine. This system of working the feed pump

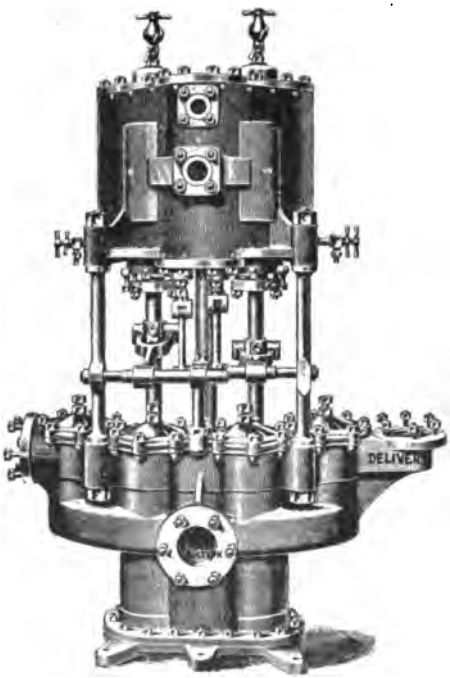


FIG. 229.—Weir Duplex Pump.

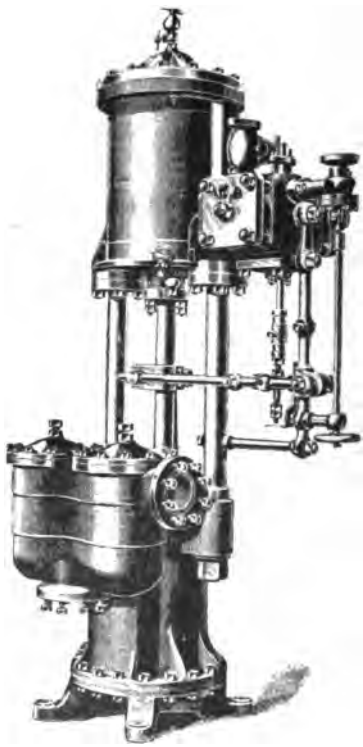


FIG. 230.—Weir Mono-type Pump.

from the main engine is still used in steamboats and other small installations.

2. The increasing rate of revolution of the main engine led to the almost general adoption of independent feed engines for both main and auxiliary use. The main feed pumps were usually situated in the engine room, and the rate of supply was regulated automatically. A cock was fitted in the steam pipe to the feed engine, and connected by a simple system of levers with a float in

the feed tank; a rise of water-level in the tank opened the cock and put the pump on faster, and a fall correspondingly eased it down. This arrangement was most satisfactory, and generally adopted for all vessels with cylindrical boilers, and is still common in the mercantile marine.

3. The rise of pressure, obtained at first by the use of water-tube boilers, and the increased number of boilers now incorporated in smaller units of the total power, led to the removal of the main feed pumps to the boiler rooms. The work done by the pumps is thus decreased with the shorter delivery pipes and consequent less frictional resistance; and the control of the feed water is under the supervision of the boiler-room staff, in its own compartment.

The independent pumps were generally of the duplex type—that is to say, there were a pair of cylinders and pump chambers to each pump. In a duplex engine the usual cranks and eccentrics are generally replaced by a pair of single levers, worked from the piston rods, each of which actuates the opposite slide valve—as in Fig. 229, which shows a Weir duplex pump. For some time past the duplex system has given way to the single-cylinder variety of direct-acting pump (Fig. 230); but the arrangement of pump valves is somewhat similar, although improved in detail.

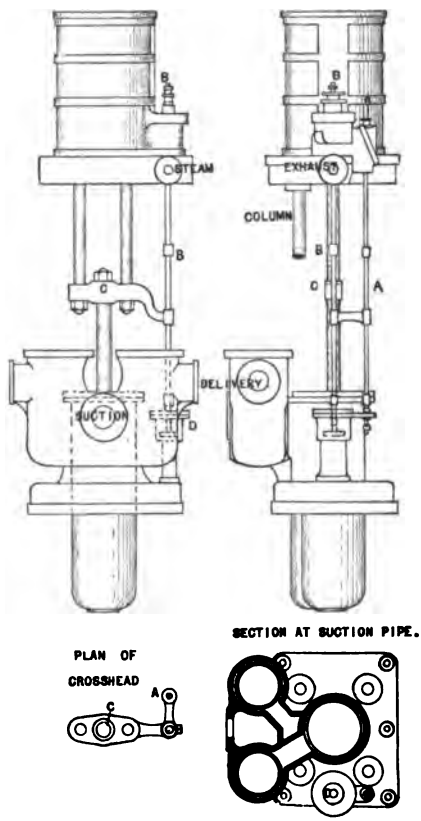


FIG. 231.—Karoom Pump.

Detail of Feed Pump.—A type of feed pump, made by the Pulsometer Engineering Company, is shown in Figs. 231 and 232. Fig. 231 shows a general outline of the arrangement of the steam and water cylinders with the rods, in front elevation; the same in side elevation; plan of crosshead, showing the relative positions of

rod connections; sectional plan of pump barrel and valve boxes. Fig. 232 shows a section through the main valve, cylinder ports, and false face; a section through the auxiliary valve, and part section of main valve; a section through one valve chamber, and details of its fittings.

There is one steam cylinder, connected by four nearly upright columns with one pump cylinder. Two steel rods are fitted to the steam piston, and the outside ends are joined by a crosshead, *C*, to which the upper end of the single phosphor-bronze pump rod is attached. With this arrangement the steam piston rods can be made of steel, and the pump rod of bronze; and the working parts can be more easily examined and refitted, as either one of the steam or water pistons can be opened up without disturbing the other.

The crosshead is made with an extension which slides over two rods, one of which, *A*, moves the auxiliary slide valve, and the other, *B*, moves the main valve, but only when the admission of steam through the auxiliary valve fails to move it. Adjustable stops are fitted in correct positions near the ends of the rods, against which the crosshead comes in contact at certain parts of the stroke, and thus moves the valves. The pump does not stop when the steam admitted through the auxiliary valve fails to move the main valve; but the engine and pump become noisy and jerky in their movements, and attract attention. The main slide valve is of the piston type, and is packed with two outer rings of cast iron and an inner one of phosphor bronze. Steam is admitted to the auxiliary valve through a small hole drilled into the main steam supply, and the exhaust is connected with the main valve exhaust in the same way.

At the lower end of the main valve rod a dash-pot, *D*, or cushioning cylinder, is fitted. This is air-tight, and the piston, working in the cylinder and attached to the rod, passes a single annular series of small holes at about half-stroke; the air above or below the piston, as the case may be, is imprisoned and compressed between the piston and end of the cylinder, and thus cushions the movement of the main valve at each stroke.

The pump chamber is fitted with two valve chests, with separate sets of suction and delivery valves in connection with each end. The valves are fitted in groups, eighteen or more in each, machined from solid rods of phosphor bronze, with coned seatings, and fitting

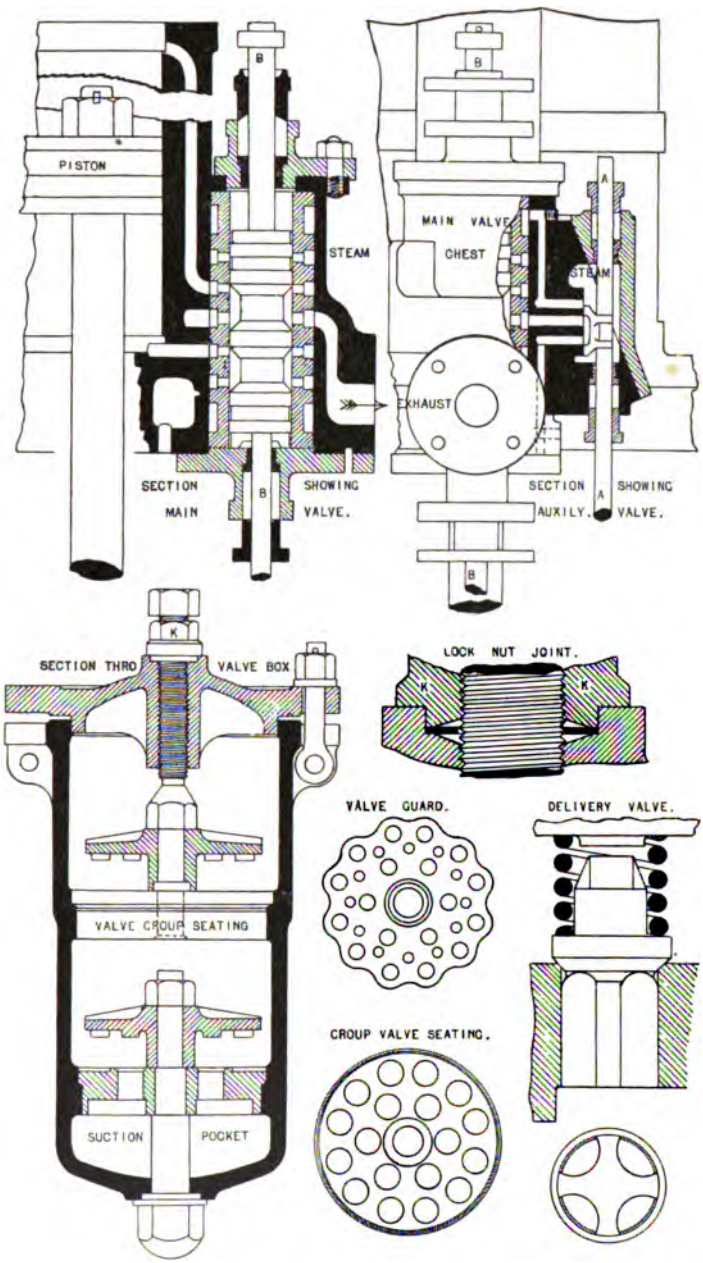


FIG. 232.—Details of Karoom Feed Engine and Pump.

into holes suitably made in the valve group seating. The group seatings are made of phosphor bronze, and are removable. When in place, they are pressure-tight on coned seatings made in the barrel of the pump. The lower, or suction, seating is held down by a stud fitted up through the bottom of the chamber; this stud also supports the guard which regulates the lift of the small valves. The upper, or delivery, seating is pressed down on its seating by a set screw through the top of the cover, and butting on top of a capped nut on the stud supporting the valve guard. The small valves are allowed a very small lift—about $\frac{3}{32}$-inch for valves about 1 inch diameter. A light gun-metal spring is fitted above each delivery valve to return it to its seating.

The cover joint is made with either gutta-percha, or with a lead ring of about $\frac{1}{4}$-inch sectional diameter, seated in opposite right-angled grooves, as shown, cut in the flange and cover. Steel bolts, hinged to the valve chest, are used to secure the cover with nuts and split pins, and with loose wrought-iron washers under the nuts. The lock nut *K*, on the set screw fitted in the centre of the cover, is jointed, as shown in the enlarged figure, with a lead washer compressed between the face of the nut (which is shaped like a flat seated valve) and the cover.

Bradford Feed Pump.—The Bradford feed engine is distinctive in that there is neither a shuttle valve nor an auxiliary valve. The slide valve is given an intermittent rocking motion, derived from the crosshead when near the ends of its stroke. As shown in Fig. 232A, the piston has nearly completed its upward stroke and is rotating the valve with an anti-clockwise motion, so as to connect the bottom of the cylinder with the central or exhaust port, and at the same time to open the right-hand port to supply the top of the cylinder with steam.

The pump-barrel is lined with gun metal, and the bucket is of the same material and fitted with packing. The essential difference between the action of this pump and others of a general similarity of type is that holes are drilled in the liner-barrel so that they are just overrun by the bucket, and they thus relieve the delivery pressure in the clearance space for a moment at the reversal of the motion of the bucket. This arrangement allows time for the delivery valves to seat themselves without shock. The pump valves are usually of the flat gun-metal disc type, which is not generally

suitable for delivery pressures of over 180 lb. per square inch; but, of course, valves of the type shown in Fig. 232 can be provided for in cases of higher pressures.

Feed Heaters.—The object of heating the feed water is to promote economy by utilising some of the heat otherwise wasted in the condenser, and thus reducing the work of the boiler. The following may be considered the advantages of feed heating:—

1. Economy, which may be anything between 1 and 20 per cent, according to the efficiency of the system employed. (The Belleville economiser is probably one of the most efficient types of feed heater; but the feed heater, as generally understood, derives its heat from steam, not from the waste products of combustion.)

2. More even temperature in the boiler, and a consequent saving from the racking strains to which it is subject, especially when raising steam and cleaning the fires.

3. Reduction of priming, and a better quality of steam generated in the boiler; it is in this way that the Belleville economiser effects part of its economy.

4. Steam can be more easily generated with hot feed water, and the heating surface appears to be more efficient.

Steam feed heaters may be divided into two classes: the injection type, which can only be used

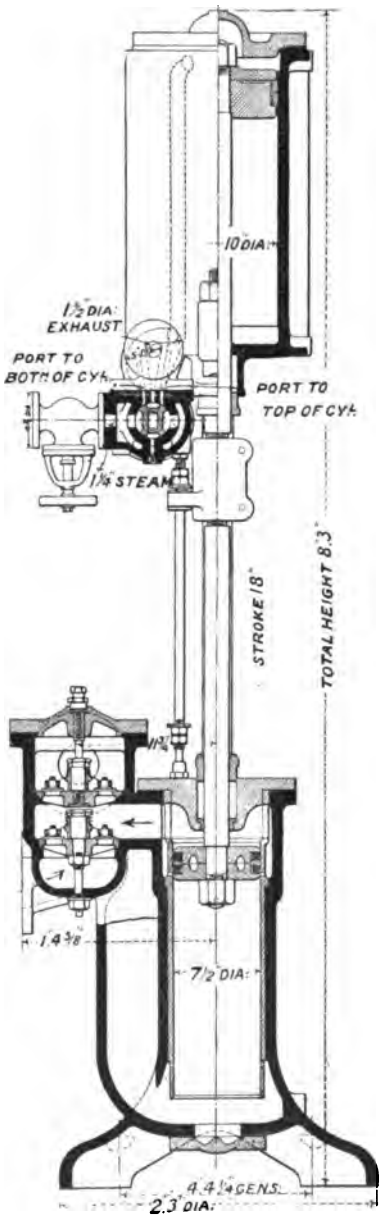


FIG. 232A.—Bradford Feed Pump.

on the suction side of the feed pump; and the surface type, which can be used on either the suction or delivery side.

With *jet injection feed heaters* some of the exhaust steam from any convenient part enters the feed tank by a pipe fitted with some kind of mixing apparatus, to allow the steam to condense and mix with the water without noise or inconvenience from local heating. The heating steam is usually obtained from the coil drain or gained steam of the evaporator, from the general steam drain system, from the auxiliary exhaust pipe, or from the receivers of the main engines (the L.P. preferably), where it has already performed some, if not the whole of the work that can otherwise be obtained from it.

The condensing of the steam amongst the feed water raises the temperature of the latter, and the limit of a jet heater is when the temperature rises to about 180°; above this limit the feed pumps are troublesome, owing to the presence of vapour and air (unless care is taken to allow them to escape) causing irregular working by their expansion when the pump draws in a fresh supply of water. All pumps work best when the water is freed from air and vapour, and, when there are any air pockets, it is good practice to connect them with the exhaust steam pipe by a small pipe and cock, and thus allow the air and uncondensable vapour to escape without losing the water carried over with them.

With *surface feed heaters* a much greater temperature can be imparted to the water by fitting them between the feed pump and the boiler, and when steam is used for heating, this temperature is only limited by that of the heating steam. The supply of heat may be taken from any of the parts mentioned for jet heaters, and in many instances is now taken directly from the boiler itself.

Live-steam Feed Heater.—When the feed water is heated by steam taken directly from the boiler, without doing other work on its passage, it is termed a live-steam heater.

Two views may be taken of its action in effecting economy—either that it is conducive to the production of a better quality of steam in the boiler, and this promotes economy in the engine; or, that the proportion of the steam used in heating the feed water is more reproductive when used in this way.

In the first case, in some particular instances, no economy could be produced by its action if the steam generated without the heater were of the highest quality, that is, perfectly saturated and

dry. Some experiments tend to prove this contention, and that practically no economy was effected by feed-heating as demonstrated by them. There is, however, a mass of evidence to prove that economy has been obtained in a great number of other cases.

In the second case, the following considerations are probable. The steam used in the heater takes the heat from the boiler only to return a large percentage of it, for some must be lost by radiation without doing any apparent useful work. It cannot return more heat to the boiler than it takes from it. The economical gain is due to the more useful manner in which it is applied; thus in heating the feed water, some 90 per cent of its heat is returned to the boiler as useful work, whereas when used in the engine only about 12 to 15 per cent of its heat is usefully employed in driving the pistons, because the greater part of the remaining heat is pumped overboard with the circulating water. The relative loss is only about one-eighth of the heat rejected in the condenser, and the economy is in the ratio which this amount bears to the total steam used. The quantity cannot be very great, because there is always a relative loss of work on the pistons and a corresponding loss of efficiency of the steam.

In any feed heater the maximum temperature to which the water can be raised is the boiling-point, due to the pressure at which the feed water is delivered to the boiler. It remains as water in the heater, and this is, therefore, all *sensible heat.* The quantity of heat imparted to the feed water must therefore be limited to that required for the attainment of this temperature. It may be considered that the fraction, or ratio, of the steam which can be used for heating the feed water is definitely governed by the quantity of sensible heat *divided* by the total heat of formation of 1 lb. of boiler steam.

For example, consider a boiler working at 225 lb. absolute pressure, with a corresponding steam temperature of 365° and a hot-well temperature of 120°; then the latent heat is 836 b.t. units, and the sensible heat, $365 - 120 = 245$ units, and the ratio is—

$$\frac{245}{836 + 245} = \frac{245}{1081} = 22{\cdot}6 \text{ per cent.}$$

This ratio of about 22 per cent may be considered a maximum attainable in ordinary practice, and the actual economy must be somewhat below it when live steam is used as the heating medium,

because there must be deducted from it some loss due to radiation and other causes. The feed pumps are also burdened with rather more duty, but the cost is very slight, probably not greater than $\frac{1}{2}$ per cent of the total power generated.

Some economy is also perhaps obtained by separating the air from the feed water during the heating process, and when delivered into the boiler, this air materially assists in the formation and liberation of the steam from the surface of the water; it also promotes circulation, and, being removed from the water near the boiler heating surfaces, allows the water to take up heat more readily.

Surface Feed Heater.—Brooke's annular film feed heater for using live steam is shown in Fig. 233. Two sizes of straight tubes are used, the smaller size passing through the larger, leaving a thin annular space or film of water between them. The steam is admitted to the exterior of the larger tubes, and, after passing upwards, turns downwards, through the interiors of the smaller tubes, to the bottom of the chamber. The water condensed from the heating steam is drained away through a steam drain trap to the feed tank or condenser. The feed water enters near the bottom of the chamber, where the inlet is separated from the outlet by a division plate, outside the smaller tubes; passes upwards in thin films between the outer and inner tubes, and turns downwards between the smaller and larger tubes of the remaining part to the outlet, which is shown opposite the inlet.

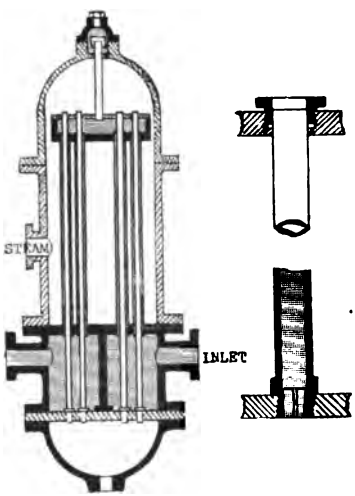


FIG. 233.—Brooke's Annular Film Feed Heater.

There are four tube plates, two each for the larger and smaller tubes. The larger tubes are expanded at each end to make tight joints with the tube plates. The smaller tubes are screwed into the bottom tube plate with cone-seated joints, and can be easily removed by a rod with a square end fitting into a socket in the end near the cone. The upper ends pass through the uppermost tube plate, and are made tight by screwed glands; this allows freedom

for expansion. The two upper tube plates are jointed together, enclosing a space through which the feed water passes, and this chamber, or floating head, not being attached to the casing, allows freedom of expansion for the larger tubes. Scum can be drawn off from the floating head through the pipe shown. The two lower tube plates contain the feed water between them, and separate it from the incoming steam above and the outgoing steam drain water below.

The surfaces of the tubes can be scraped or cleaned by removing the top cover and smaller tubes; this is one of the advantages claimed for this type.

CHAPTER XX

EVAPORATORS AND DISTILLERS

General Principles.—The evaporator is used to obtain fresh water from sea water, and thus to make up the loss from leakage of the boiler feed water, which, as already pointed out, should be fresh water when used for high-pressure boilers. Water must also be supplied for drinking, cooking, and other purposes; this is also obtained by using an evaporator, and, in conjunction with it, a condensing and cooling apparatus, called the distilling condenser.

By using an evaporator the scale, otherwise deposited on the boiler heating surface, is deposited in the evaporator, principally on the heating coils; the boiler heating surfaces can therefore be kept free from scale and in a high state of efficiency for long periods.

The evaporator is now generally vertically arranged, as shown in Fig. 234 (*Kirkaldy*), as it then takes up less ground space. The chamber is nearly cylindrical for a great part of its height, and in its lower part coils of copper piping are fitted, which are nearly immersed in sea-water, admitted, or pumped, into the evaporator as required. *Primary steam* is admitted to the coils internally, and the sea-water is boiled and *gained steam* is produced by the heat thus imparted; while at the same time the steam inside the coils is condensed and continuously drained away through the *coil drain* to allow fresh steam to take its place.

The *capacity of an evaporator* is measured by the amount of heating or coil surface it contains; but its rate of production (which might also be considered its capacity) depends on the difference in temperature between the heating or generating steam and the generated or gained steam, and on the cleanliness of the heating coils, both internally and externally.

The same principle is used in all evaporators, and it is only the difference of arrangement of the details that constitutes any difference in working. The mountings are similar to those on a boiler, in which the heating coils take the place of the furnace. Sea-water is admitted nearly up to the top level of the tubes, as shown in Fig. 234; steam is admitted inside the coils, and heats

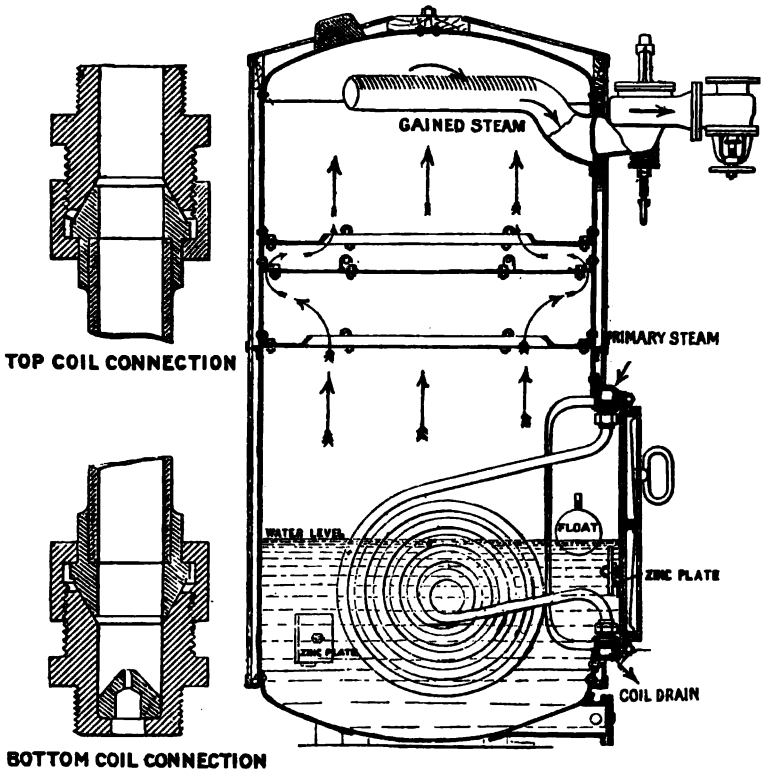


FIG. 234.—Section through (Kirkaldy) Evaporator.

the water surrounding them, and finally ebullition takes place. The steam thus generated is called the *gained or secondary steam*; and the steam used inside the coils, which may be taken either from the boiler direct, or from the main engine receiver, or from the closed exhaust system, is called the *primary or generating steam.* The gained steam is taken away from the top of the chamber through a valve and pipe, either connected with the condenser, or, as in a few instances, with the L.P. receiver, where it supplements

the exhaust steam from the M.P. cylinder. The water condensed in the coils, by parting with its heat in generating the gained steam, is led away by a drain-pipe, called the *coil drain*, into the condenser, feed tank, or feed heater; whence it is returned to the boiler, with practically little loss from leakage. When the evaporator is used for making water for drinking purposes, the gained steam is taken to a separate condenser, called the *distilling condenser*, from which it runs or is pumped into the ship's tanks set apart for the purpose.

Evaporator Fittings and Mountings. — The fittings and mountings on an evaporator are very similar to those fitted to a boiler, and are for very similar usage. They are generally as follows:—

FIGS. 235 TO 239

A, Safety valve for generated or gained steam.
B, Pressure gauge for gained steam.
C, Outlet steam control valve, or vapour control valve.
D, Outlet stop valve.
E, Inlet for low-pressure steam to coils, non-return valve.
F, Inlet for high-pressure steam to coils, generally attached to *E*.
G, Drain valve from coils, conveying away water condensed from the steam in the coils.
H, Safety valve in connection with coils.
J, Pressure gauge in connection with coils, primary steam.
K, Water gauge.
L, Blow-out valve connected with the sea.
M, Blow-out valve connected with the bilge.
N, Feed inlet and regulator. (Inlet to regulator fitted with a sentinel or safety valve.)
O, Brine mixing and cooling chamber (Fig. 238).
P, Cooling water inlet to brine chamber from feed-supply pipe.
Q, Brine suction from cooling chamber to brine pump.
R, Hydrometer fittings for testing density.
S, Mushroom cap separator plate.
T, Return pipe for separated water.
Z, Portable stand-pipe, in connection with coils.

Weir L.P. Evaporator.—In this arrangement the upper part of the chamber is cylindrical, and made of steel plates, with a dome-shaped top. The lower part of the chamber is flat-sided, made of

cast gun-metal generally, and connected with the upper part by a bottle neck and flanges. The upcast passage from the lower part

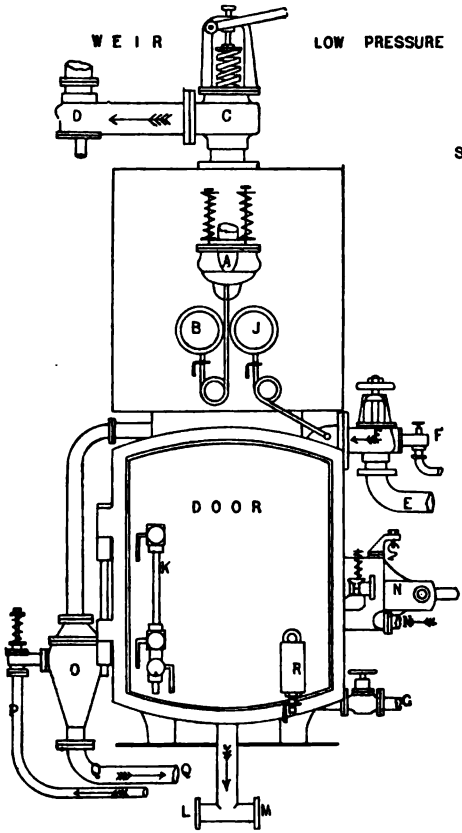


FIG. 235.

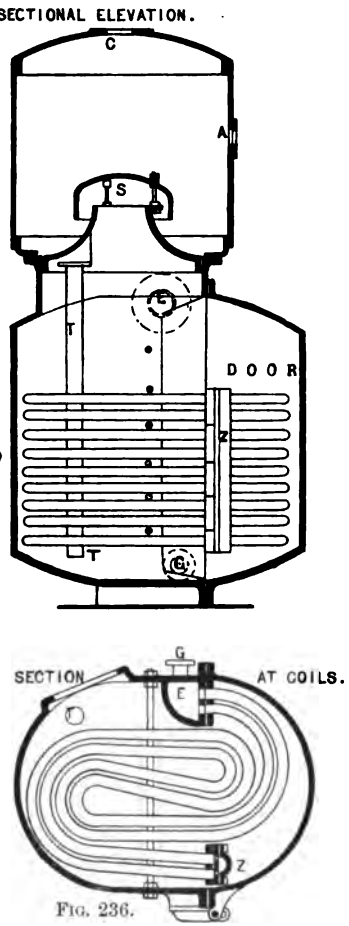


FIG. 236.

FIGS. 235 and 236.—Weir L.P. Evaporator.

extends into the upper part, with a very much reduced area, and above it a mushroom-shaped cap, *S*, is fitted to act as a separator and to deflect the upcast steam and any water carried with it; the

water is thrown down, and drains back to the bottom of the chamber through a stand-pipe, *T*. The lighter steam rises and passes away, through the vapour valve *C* fitted on top of the chamber, to the condenser or distiller.

The solid drawn copper tubes or coils (Fig. 236) are fitted in pairs horizontally above each other. Steam is admitted to the first two pairs at one end from the inlet pipe *E*, passes through the four coils, and escapes through a small hole at the outlet end, *Z*, of each. The hole is generally about $\frac{1}{4}$-inch diameter for a coil about $\frac{3}{4}$-inch internal diameter. Thence the steam passes through unrestricted openings into a fresh set of coils or element. The outlet passages are again restricted, and so on until the last pair of coils at the bottom is reached, when the outlets are unrestricted and the water is drained away through the coil drain valve into the condenser. Each coil is connected with the vertical inlet and outlet pipe *E* and *Z*, and webs are cast in them to separate the inlet and outlet openings.

The ends of the tubes forming the coils are expanded into flanges, which are bolted at one end to the steam inlet pocket at *E*, and at the other to a portable stand-pipe, *Z*. This stand pipe is made to restrict the passage outlets as required. In a later type of fitting the tubes are fitted to the stand pipe and inlet pocket by ground couplings in place of flanges, each tube separately.

The lower part of the chamber is fitted with an opening and hinged door for access to the heating coils. The nuts are first removed from the studs, and then the door pulled back until the door can be swung open clear of the studs, after which the coils can be examined or removed for cleaning. Another door and opening is frequently fitted at the back or sides for removing any dirt or scale without first removing the coil.

Evaporator Feed and Brine Pumps.—If the gained steam is maintained at sufficient pressure, the brine can be blown out in the ordinary way; but it is more usual to work evaporators at very low pressures, and frequently below that of the atmosphere. Consequently the brine must be withdrawn, and a pump, generally worked from the same piston as the feed pump, is used to withdraw it by suction. A pump is very irregular in its motion when drawing hot water from a vessel in which the pressure is below that of the atmosphere, and a more uniform motion is obtained by cooling the brine before it is withdrawn.

In the Weir arrangement (Fig. 237) the feed and brine plungers, *A* and *B* respectively, are attached to the same rod, *C*, which also forms the steam piston rod. Each plunger has its own set of valves, *D* and *E*; the feed pump has twice in the older patterns, and about $1\frac{2}{3}$ times in the newer patterns, the capacity of the brine pump. If the quantity of feed water supplied to the evaporator is twice that of the brine withdrawn

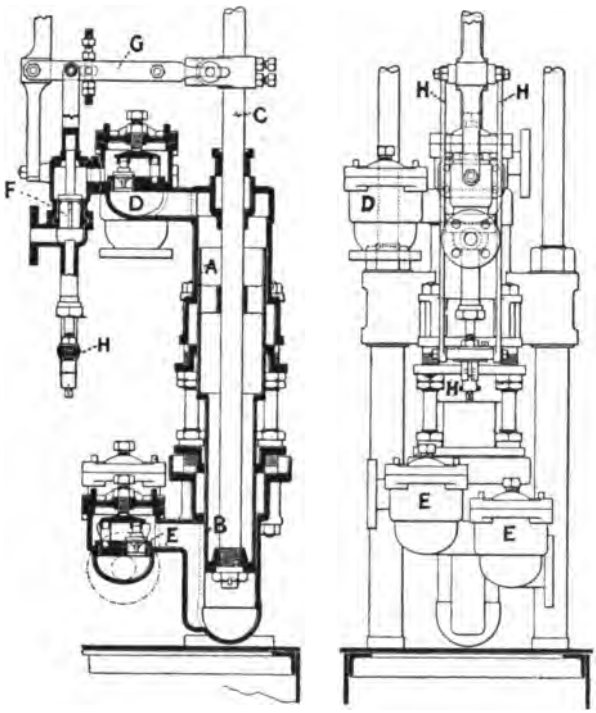


FIG. 237.—Evaporator Feed and Brine Pumps.

by the brine pump, a little consideration shows that the density of the water in the evaporator is maintained steadily at twice that of the feed water, which for ordinary sea-water feed is evidently 20° or $\frac{2}{3}2$. In practice the ratio of 2 to 1 is raised by a portion of about one-fifth of the feed water being used to cool and dilute the brine; consequently the brine pump must be of slightly greater capacity to maintain a steady density of 20°.

The capacity ratio and consequent density are independent of the rate of motion of the pump rod, and to provide for variation in

working conditions the quantity of cooling water can be regulated by a valve, *F*, the spindle of which is connected with the slide valve actuating lever *G*, and with an adjustable brine index, *H*. By altering the index the instant of opening of the valve *F* in relation to the total plunger stroke can be fixed for any desired working density. For a typical Weir arrangement, the densities, etc. are shown in the following table :—

Valve opens for Supply of Cooling Water.	Total Feed.	Cooling Water.	Brine.	Ratio Brine to Total Feed.	Ratio Cooling Water to Brine.	Equivalent Density in Evaporator.
At ·394 Stroke	·394	·606	0	No Brine	No Brine	Increasing Density
,, ·4925 ,,	·4925	·5075	·0985	$\frac{1}{5}$	5·15 / 1	$\frac{5}{32}$
,, ·591 ,,	·591	·409	·197	$\frac{1}{3}$	2·08 / 1	$\frac{3}{32}$
,, ·656 ,,	·656	·344	·262	$\frac{2}{5}$	1·31 / 1	$\frac{2\frac{1}{2}}{32}$
,, ·788 ,,	·788	·212	·394	$\frac{1}{2}$	1 / 1·86	$\frac{2}{32}$
,, ·985 ,,	·985	·015	·591	$\frac{3}{5}$	1 / 39·4	$\frac{1\frac{3}{8}}{32}$

Relative Volume of Feed Pump = 1
Relative Volume of Brine Pump = ·606

The *brine cooler* is fitted to the side and near the bottom of the evaporator. It is connected at the top 3 with the steam space, and at the side 1 with the water space, of the evaporator. The cooling water enters near the top 2 through a spring-loaded valve (loaded to prevent sea-water being drawn in when working the evaporator below atmospheric pressure), and is deflected downwards by an internal pipe in connection with the top pipe. The brine flows in at the opposite side 1 of the internal pipe, and in falling to the bottom mixes with the cooling water, whence the mixture is withdrawn from the bottom 4 of the cooler by the brine pump.

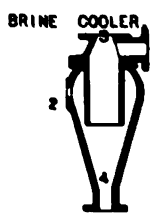


FIG. 238. Brine Cooler.

The cooler, brine pump connections, and valve require cleaning frequently, and it is good practice to do this every time the coils are cleaned or scaled. The addition, below the cooler, of a sediment box, which can be readily cleared, considerably reduces the deposit of scale and consequent choking of the brine pipe and valve connections.

Feed Regulator.—The feed control box is generally attached to the outside of the evaporator, and is then readily accessible. Inside the box a copper float, tinned all over, is connected, through an internal lever, 8, and spindle, with an external lever, 7, which is fitted with a balance weight or a spring in tension (Fig. 239).

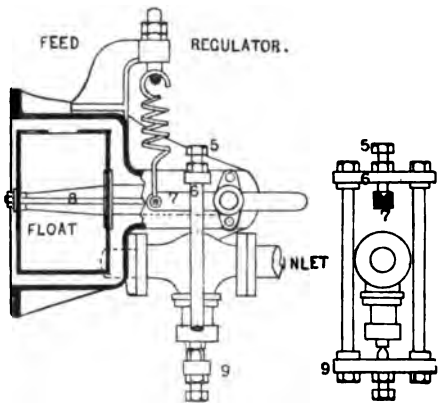


FIG. 239.—Feed Regulator.

When the float is full of water it balances the external weight or spring, and a cock is now usually fitted to fill the float with water, when required. The float regulates the amount of feed water through the external lever, which works the regulating valve through the adjustable frame 569, and when the working level is normal a pressure of about 50 lb. per square inch is maintained in the feed pipe by the spring tending to close the valve; the original balance of the spring is upset when the float is partially water-borne by the water displaced by it.

The efficient working of the feed regulator depends on the accurate fitting of the valve and on the packing in the rod and valve-spindle glands. Soft cotton packing, well impregnated with black-lead, is recommended by the makers, to avoid undue friction.

A *sentinel valve*, loaded to about 80 lb. pressure per square inch, is fitted on the delivery pipe from the pump, and if this valve lifts, the pump should be eased down; there is a tendency to maintain a higher working level when the pressure is too high, because the pressure on the valve tends to overcome the tension exerted by the spring.

Starting and Working the Evaporator.—All evaporators prime readily, from the same causes which induce priming in a boiler. Irregular working, leakage of the internal feed pipe above the working level, the impact of cold air on the outside of the steam space when imperfectly lagged, and the low pressure at which the steam is generated, are the principal causes in ordinary practice.

The necessary openings of the valves and pressures to obtain the full output of the evaporator are generally shown on an in-

struction plate attached to the front. The working level is generally at or below the upper tubes, and if the water gauge is correctly fitted it should then show nearly a full glass.

When starting, it is advisable to proceed in the following order :—

Adjust the brine index, if required.

Start the pump and fill the evaporator until the feed regulator exerts a pressure.

Open the coil drain, generally *full open* when working.

Admit steam to the coils *slowly*, and allow the pressure to rise inside the evaporator steam space to 2 or 3 lb. above the atmospheric; then very gradually open the vapour outlet valve until the requisite pressure or vacuum is obtained.

The guaranteed output, when clean, can be obtained either with a pressure of about 6 lb. in the coils, and 9 inches of vacuum in the steam space, or with 16 lb. in the coils and 1 to 2 lb. pressure in the steam space.

Test the density every hour, and see that it agrees with the brine index. If it does not agree, there is generally some unnecessary leakage.

The evaporator should be blown down every twelve hours. To do this, close the vapour valve and allow the pressure to rise to about 12 to 15 lb. per square inch; then open the blow-down quickly to the sea, and as soon as the water is entirely blown out, close the inlet steam valve to the coils. The entrance of the cold feed water creates a vacuum, and water is drawn in quickly through the blow-out, which should be closed as soon as the water begins to show in the gauge glass, when the inlet can be opened and the working restarted.

Vapour Control Valve.—Fig. 240 shows a cross section through a Kirkaldy vapour control valve. There is a cylindrical sliding valve working inside a cylinder. Below, the valve is connected with the steam space of the evaporator through an internal steam pipe. Above, the valve is open to the atmosphere which tends to force the valve downwards and so cover the outlet ports. The valve is packed so as to ensure a good working fit in the barrel; and a ring face is formed on its upper surface which, when the valve is fully open, makes a steam-tight joint with a seating on the cover. A spring, in compression, tends to keep the valve open in opposition to the atmospheric pressure, and by

adjusting the compression of the spring the valve can be set so as to maintain a constant pressure or vacuum in the evaporator steam space. In the figure the valve is shown fully open, and the spring set for 8 inches vacuum by means of the check nuts. Messrs. Kirkaldy do not recommend a vacuum greater than 5 inches; and in the author's experience the best average results are obtained when evaporating at atmospheric pressure, using a pressure of 20 lb. per square inch in the coils supplied by the closed exhaust. From various causes, principally due to fluctuations in pressure of the closed exhaust and in the condenser, and to fluctuations of movement of the steam and vapour in the respective pipe connections, the ideal conditions of a high vacuum and low coil-supply pressure cannot be realised in ship practice; but in shore practice, where a steady pressure is assured at all times, better average results can be obtained.

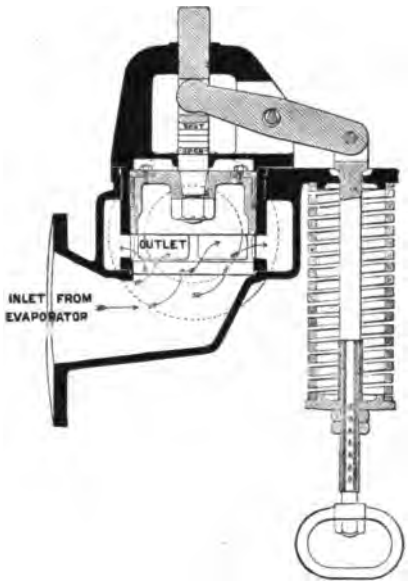


FIG. 240.—Vapour Control Valve.

Cleaning.—The rate at which steam is generated in an evaporator is due to the rate of transmission of heat from the generating steam to the water surrounding the coils, which rate is dependent on the difference of temperature between the generating and generated steam and the cleanliness of the heating surfaces, both internally and externally.

When the generating steam is taken directly from the boiler, there is only a very slight deposit of grease inside the tubes of the coils; but when it is taken from the engine receivers, or from the closed exhaust system, the deposit may seriously decrease the efficiency. The amount of the deposit is intimately connected with the internal lubrication of the engines; and when boiler steam is used, on the efficiency of the grease extractor. Many engineers use no internal lubrication for either main or auxiliary engines,

and very few use any for the auxiliary engine cylinders; but a small quantity of greasy matter finds its way into the steam from the piston rods and valve spindles. For good practice, such as the above, the internal surfaces of the tubes may require cleaning after about six months' total working; if the external surfaces are clean, the amount of the internal deposit is evident from the falling off in the rate of production.

The internal surface of the tubes can be best cleaned by scalding them through with a strong solution of caustic potash or soda, and then cleaning them with a wire-handled brush and clean water. Generally the coils must be removed from the evaporator for this purpose, and in some cases they can be placed over a slow fire and the grease burnt off; but care must be taken not to weaken the metal of the tubes when doing this.

The external surfaces of the tubes should be scaled with a blunt instrument after about ten days' working if steam is generated below the atmospheric pressure, or after about seven to eight days if steam is generated above the atmospheric pressure. Doors are provided for access to the coils for temporary scaling, but to thoroughly clean the coils it is better to remove them from the evaporator. The scale can be easily removed if the evaporator is worked up to the moment of opening up, and just previous to this the evaporator should be blown down and refilled, to soften and crack off the scale. In no case should the scale be allowed to dry and harden on the coils, because it is then very difficult to remove. In some cases the coils can be heated up when the door is open, and a fire-hose can be used to suddenly cool them, and some scale cracked off. Care should be taken not to roughen the surface of the tubes when scaling, because any roughness increases the difficulty of detaching scale in the next cleaning.

It should be possible to thoroughly clean the heating coils and internal parts of the evaporator in one working day. A scale should be allowed to remain on the upper parts of the steam space to protect it from corrosion; but if this scale is loose, it should be removed and a thin wash of Portland cement brushed over it before closing up. Two or three coats are generally applied, and the wash is preferably mixed with sea-water. Zinc protectors are generally fitted in the steam space, and sometimes in the lower part near the coils; but the latter is now sometimes unnecessary, because the coils are made of copper and the casing of gun metal.

After replacing the coils, the joints should be tested by turning on steam. All well-arranged evaporators allow this to be done before replacing the door; any leakage tends to induce priming.

When perfectly clean and free from scale, an evaporator will produce about 1 ton of gained steam for each 1·1 to 1·25 tons of steam condensed in the coils; but the formation of scale gradually reduces the original efficiency of 90 or 80 per cent to about 55 or 50 per cent with about $\frac{1}{8}$-inch scale, and thus increases the steam consumption per ton of steam gained by $1\frac{3}{4}$ to 2 tons.

By increasing the pressure (and, therefore, temperature) of the generating steam, the capacity can be fairly well maintained, although the efficiency is decreased by lack of cleanliness.

Distilling Condenser.—The principle of a distilling condenser is exactly the same as that of the main or auxiliary condenser; but the distiller is usually smaller, and arranged vertically instead of horizontally, not for efficiency, but for convenience.

In Weir's arrangement of distilling condenser, the tubes are vertically arranged, as shown in Fig. 241. The circulating water passes internally through them in an upward direction, and the steam from the evaporator or boiler entering from the top, being condensed on the outer surfaces of the tubes, falls to the bottom of the chamber. The condensed water is allowed to accumulate until the level rises to that shown, when it can be either run or pumped away through an outlet fitted on the air pipe at the side of the distiller. The accumulation of water allows a much greater time for cooling it as well as condensing it, and the water being a better conductor of heat than the steam from which it is condensed, the process of cooling is accelerated to the same extent. Air is admitted through the valve shown, to a space formed between a hollow cylindrical hanging screen and the outer casing. This free admission of air balances the pressure, by preventing the formation of a vacuum from the condensation of the incoming steam with that of the outflow pipe, which is also an air pipe. The air admitted mixes to some extent with the incoming steam in the distiller, and aerates it. The water produced is cold; and the outlet for the condensed water, it will be noticed, is nearest the circulating inlet, or coldest water and surface.

The tubes are expanded to make them steam-tight in their holes, as in an ordinary boiler. Expansion of the tubes is allowed by making the top tube plate in the form of an expansion gland.

Any distorting effect produced is taken on bends provided in the pipes in connection with the steam inlet and circulating outlet. The top and bottom end plates, or covers, are stayed together by three or four long stays passing through cooling tubes. A test cock is fitted on the outlet pipe for the condensed water, and there is a drain at the bottom of the same pipe.

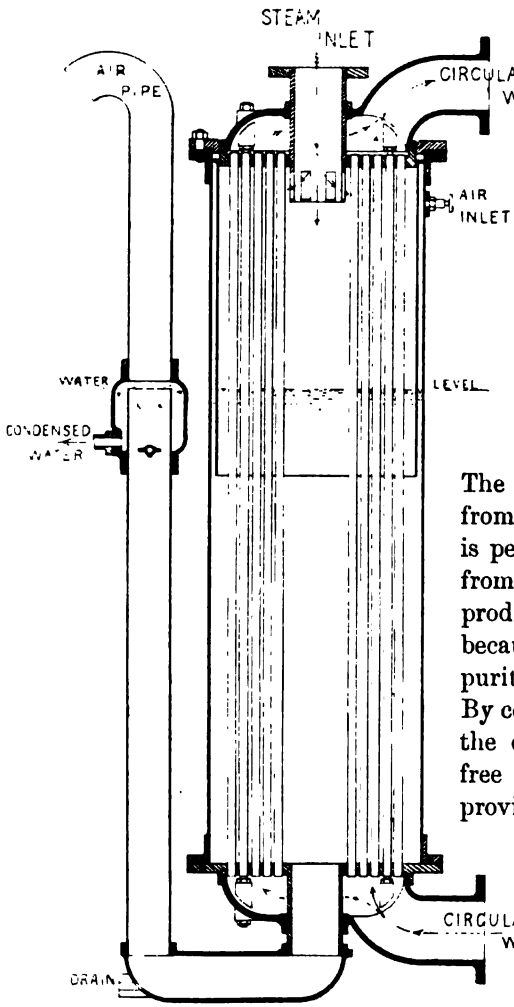


FIG. 241.—Weir's Distilling Condenser.

The water from the steam gained in the evaporator, when condensed in the main or auxiliary condenser, is of a slightly greasy nature, and not suited for drinking purposes. The sea-water in the evaporator from which the steam is produced is perfectly clean, and therefore free from grease; but the fresh water produced from it is rather tasteless, because it is freed from air and impurities in the process of boiling. By condensing it in a clean chamber, the distilling condenser, it remains free from grease, and means are provided for aerating it during condensation. In earlier times the steam condensed for drinking purposes was taken directly from the boilers, and although condensed in a separate and clean vessel, was always more or less impregnated with grease; but since the use of the evaporator the water has generally been absolutely pure and wholesome, although slightly tasteless. The water now supplied to the boilers from the feed tank compares very favourably with that which

was originally supplied for drinking, and could be used if other sources failed.

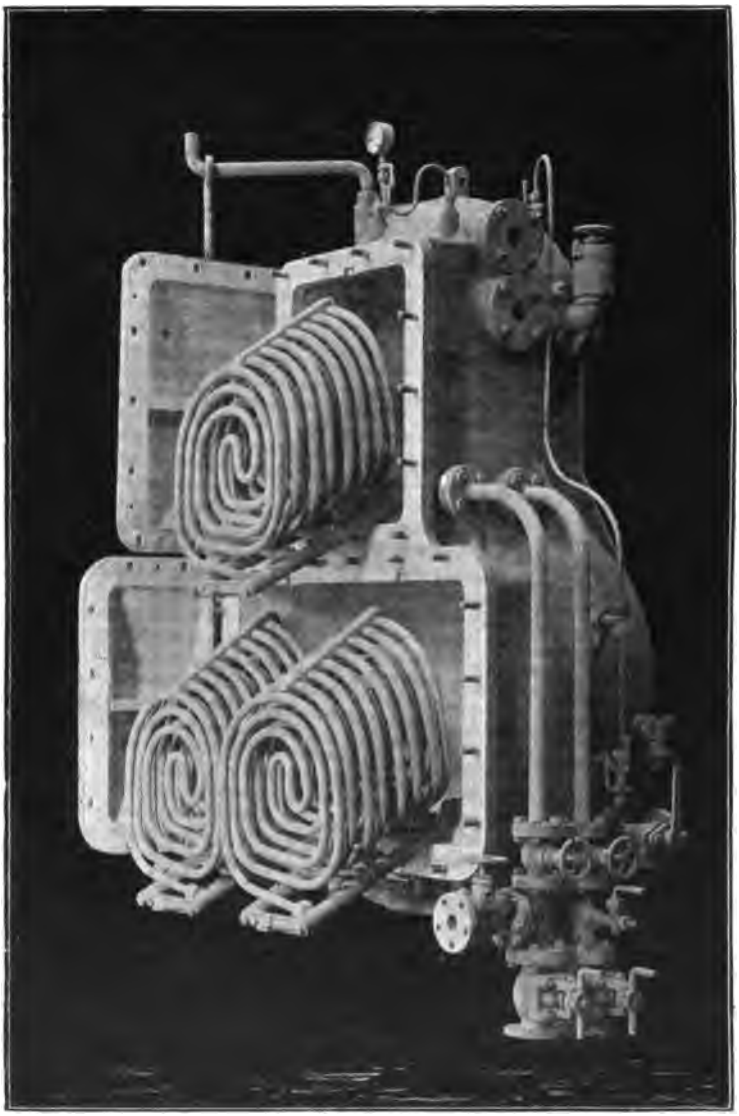


FIG. 242.—Morison's Evaporator and Winch Condenser.

Morison Radial Evaporator. — In Fig. 242 a Morison

evaporator is shown arranged for use also as an auxiliary condenser. It consists of a vertical cylindrical chamber provided with necessary formations for fitting two or three coils in its interior. The upper coil is used only when the evaporator is used as a condenser. The lower part is fitted with a swing door on a pivoted arm, for convenience of ready removal, and when open the coils are exposed. The heating coils are arranged in pairs, and are detachably connected with two tubes, forming respectively the inlet for steam and the outlet for the resulting water condensed. Each of these tubes is closed at its free end, and at its other end terminates in a cap mounted on a hollow plug, so that the tubes and the attached coils can be readily swung on the hollow plugs as an axis, and so withdrawn from the vessel for cleaning purposes, and afterwards swung back again. These operations can thus be effected without breaking any steam connections. When it is necessary to replace a set of coils with a clean set, two nuts are removed from the plugs, the inlet and outlet tubes with all the attached coils are lifted off, and the spare set lifted on and fitted up. After the coils are swung back into their working position, steam can be turned on and all the joints tested in place before the door is closed up: this is a great convenience, and any defective joints can be made good.

Kirkaldy Evaporating and Distilling Apparatus.—A section through a Kirkaldy evaporator is shown in Fig. 234; and Fig. 243 shows one complete set of two evaporators *A* and *B*, one feed-water heater *C*, one brine pump *D*, one distilling condenser *E*, and one circulating and one fresh-water pump *F* (driven by the same engine), as fitted in each of the turbine engine rooms of the *Dreadnought*. Each evaporator can be worked separately, and the gained steam either passes away through the vapour valves to the condenser, or through separate valves connected with a common pipe leading into the distilling condenser from which it is removed, as water, by the fresh-water pump. The condensed primary steam from the coil drain passes either into the feed-water heater, if required, or into the feed tank or condenser through a bye-pass pipe. The brine is removed, through a diluting box (brine cooler), from the lower part of the evaporator by the brine pump and discharged overboard. The evaporator feed water is supplied by the circulating pump; it first passes through the distiller and, becoming slightly warmed during its passage, is discharged into the bottom of the feed heater; it next passes upward through the feed heater and is discharged at

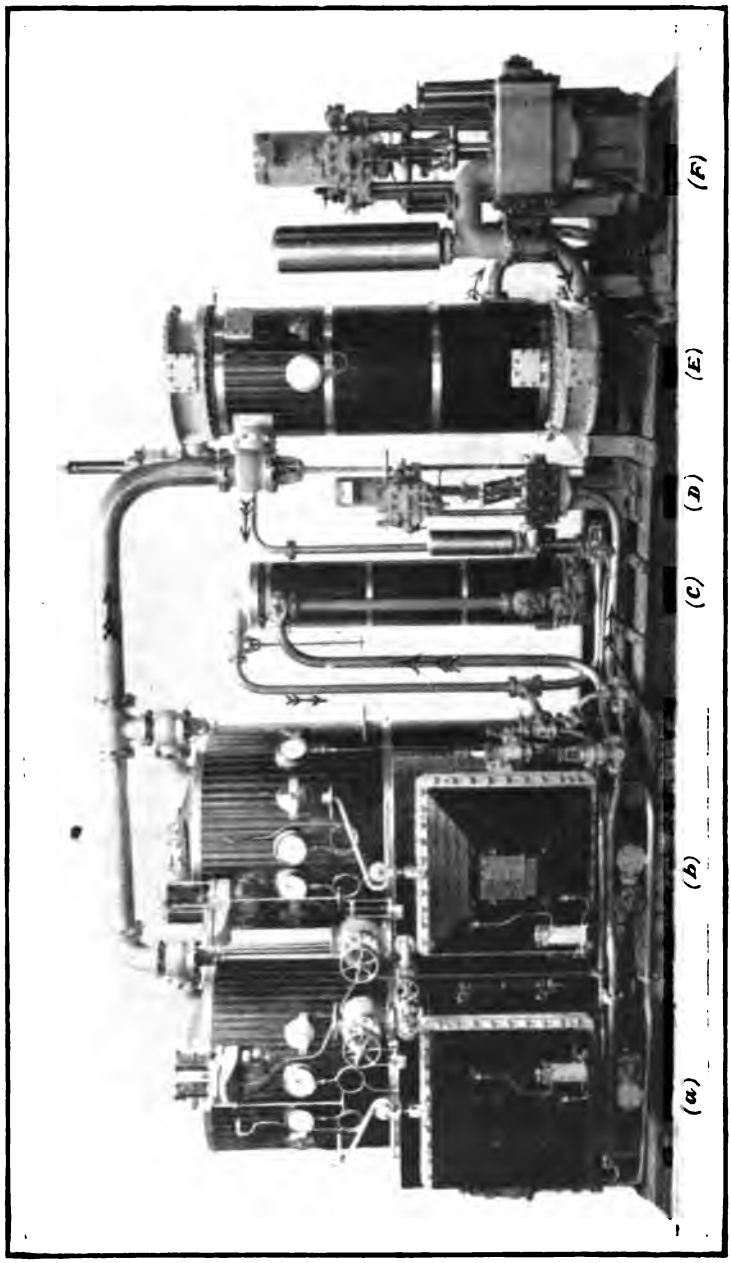


FIG. 243.—Kirkaldy Evaporator and Distilling Plant.

the top through the check valve into the evaporator feed pipe. A part only of the circulating water may be required as evaporator feed, and the surplus escapes overboard through a spring-loaded valve, shown near the top of the distiller. Each of the evaporators will produce, when clean, about 50 tons of fresh-water steam from sea-water in 24 hours' working; and the total production for the four evaporators is thus 200 tons, or nearly 45,000 gallons, in every 24 hours.

In the Kirkaldy distilling condensers and feed heaters, the coils are of elongated spiral form; some of the early evaporators were also fashioned in the same way, but the recent designs are as shown in Fig. 234. The advantage of a spiral formation is that a large area of heating surface can be put into a small space, and generally that freedom for expansion under variations of temperature which must be allowed for in some way. Figs. 241 and 233 are good examples of allowing for expansion when straight tubes are used.

Coal used for Producing Fresh Water.—Before the introduction of the evaporator, water was produced by condensing steam taken directly from the boiler, and about 7 to 8 tons of water were produced per ton of coal burnt. The scale left on the heating surfaces reduced the efficiency of the boiler considerably, and the evaporator, when using boiler steam for which an average of only 5 tons of fresh water is produced per ton of coal burnt, by maintaining the boiler efficiency, repays the consequent loss compared with direct distillation.

If the steam used for the evaporator be taken from the closed exhaust system, otherwise wasted in the condenser, the average production is increased to 10 or more tons; on the other hand, there is an increase in the wastage of boiler feed water.

There is little room for improvement in the thermal efficiency of the evaporator, because nearly 90 per cent of the heat supplied to it may be used in producing fresh water, and for any economy we must therefore look to the source from which the supply is obtained. For warships, which are distilling when in harbour in many parts of the world, where the shore supply is not above suspicion, the steam which may be supplied for heating the coils cannot be always taken from the engine receivers after doing some portion of its work and before it is rejected into the condenser. There is another system, known as the "multiple" or "double-effect" system, which appears applicable to the special case of

warships, and which has been applied with some success in some of them anterior to the application of the closed exhaust system.

The system of *double-effect evaporators* consists in fitting two evaporators in series. The gained steam in the first of the series is employed as the generating steam in the second. Practically little heat is lost in the first, and any steam gained increases the output per ton of coal burnt. From 10 to 12½ tons of water have been obtained per ton of coal in this way. The objection to the use of "double effect" is *priming* caused by the difficulty of regulating the evaporators when fitted in series; but a return to this system, with the more enlightened experience now gained with evaporators worked at low pressures and fitted with a fairly efficient control valve on the outlet pipe of the gained steam, is very likely to produce more favourable results.

For warships, under present conditions, working with the closed exhaust system, about 10 tons of water are produced per ton of coal burnt. An allowance of about 3 gallons of water is required for each person in the complement, with an extra allowance of 2 gallons for each one of the engineer staff, and 1 gallon extra for each officer, *per diem*. This works out at about 15 tons of water per day for general purposes in a battleship or large cruiser, and the boiler loss which has to be made up varies from 2½ to 5 tons per day; the cost in coal is therefore about 2 tons, or a weekly consumption of about 14 tons in harbour.

The capacity of the evaporator plant in warships is generally from five to six times the capacity of the distiller plant, and an allowance of about 10 tons per 24 hours for each 1000 I.H.P. is generally fitted for the combined output of the evaporators. A greater quantity of water could be passed through the distillers than that specified, but any increase in quantity would of course produce a corresponding increase in the temperature of the water condensed.

Other well-known makers of evaporators, distillers, and feed-heaters are *Messrs. Caird & Rayner*, the *Liverpool Engineering Co.*, *Quiggans*, *Mirlees*, *Royle* (Row's patents), and *Normandy*. The principal differences exist in the form and fitting of the heating coils, and in facilities for cleaning and repairs.

Quantity of Feed Water required to Maintain a Constant Density.—The weight of sea-water which is admitted to the boiler or evaporator as feed water is not all evaporated, because a portion

is blown out, as water, to reduce the density of the water remaining in the generating vessel.

If the brine blown out be supposed to contain n times the density of the feed water, and x = weight of steam leaving the boiler or evaporator, y = weight of water blown out; then, if the feed water contains $\frac{1}{a}$ part of salt—

$$\text{The weight of salt pumped in} = \frac{x+y}{a},$$

$$\text{and the weight of salt blown out} = \frac{n \cdot y}{a}.$$

If a constant density is maintained, the weight of salt blown out must be equal to the weight of salt pumped in, and therefore—

$$\frac{x+y}{a} = \frac{n \cdot y}{a},$$

$$\text{and } y = \frac{x}{n-1}.$$

$$\text{The weight of feed water required} = x + y = x + \frac{x}{n-1} = \frac{x \cdot n - x + x}{n-1}.$$

$$\text{The weight of feed water } (x+y) = x \cdot \frac{n}{n-1},$$

$$\text{and the weight of water which must be blown out, } y, \Big\} = \frac{x}{n-1};$$

$$\text{or, proportionately, } \frac{y}{x} = \frac{1}{n-1}.$$

For example, if $n = 2$, then $y = x$
$n = 3$ „ $y = \frac{1}{2}x$
$n = 4$ „ $y = \frac{1}{3}x$, and so on.

And if $n = 2$, the quantity of feed water required $= 2x$
$n = 3$ „ „ „ $= \frac{3}{2}x$
$n = 4$ „ „ „ $= \frac{4}{3}x$
$n = 5$ „ „ „ $= \frac{5}{4}x$.

Heat Lost by Blowing Down.—The least scale is deposited in an evaporator when the density is kept at about $\frac{2}{32}$, or 20°, by the

service hydrometer, and, from the previous section, the quantity of feed water required is then about twice the amount of water evaporated. The amount of heat carried away through brining down can be calculated as below—

Let x = quantity of water evaporated during any given time;
y = quantity of water blown out during the same time;
t = temperature of water pumped in as feed;
T = temperature of steam generated.

Then, because the brine is blown out as water at the temperature T—

$$\text{The loss of heat} = y \,.\, (\mathrm{T} - t).$$

The heat expended usefully, that is, in evaporating water—

$$= x \,.\, \{966 + 0{\cdot}3(\mathrm{T} - t)\};$$

and, expressed as a fraction, the heat loss—

$$= \frac{y \,.\, (\mathrm{T} - t)}{x \,.\, \{966 + 0{\cdot}3(\mathrm{T} - t)\} + y\,(\mathrm{T} - t)}.$$

If the density is maintained at n times that of the feed water—

$$\frac{y}{x} = \frac{1}{n - 1},$$

and therefore the loss

$$= \frac{\mathrm{T} - t}{966 \,.\, n + (\mathrm{T} - t)\,(0{\cdot}3n + 0{\cdot}7)}.$$

EXAMPLE.—(1) If n is four times the density of sea-water, and the feed water be of the same density as sea-water, as when using a jet condenser, and $t = 60°$, T = 300°—

$$\text{Then the loss} = \frac{300 - 60}{4 \,.\, (966 + 114)} = 0{\cdot}05\dot{5}, \text{ or not quite 6 per cent.}$$

EXAMPLE.—(2) If $n = 2$, $t = 80$, and T = 220, as when using an evaporator—

$$\text{Then the loss} = \frac{220 - 80}{2 \,.\, (966 + 91)} = 0{\cdot}066, \text{ or not quite 7 per cent.}$$

PART VI

STEAM—THE WORKING SUBSTANCE

CHAPTER XXI

THE INDICATOR AND INDICATOR DIAGRAMS

THE indicator is used to obtain a diagram showing the pressure at each instant of the stroke of the piston in a steam, or other fluid, cylinder; and from this diagram much other useful information is obtained. By the proper use of the indicator the mean pressure on each side of the piston can be obtained, and, when the other details are known, the indicated horse-power developed in the cylinder can be calculated.

The action is similar in nearly all indicators; but the motion of the pencil, used to mark the pressure on the diagram paper, is obtained by many different applications of the principle of parallel motion, or pantograph.

M'Innes-Dobbie Indicator.—There is a light cylindrical metal barrel, *O*, round which a paper is stretched, as shown in Fig. 246. The barrel is moved round on its axis by a string, *L*, which is kept in tension by a spiral spring inside the barrel. The other end of the string is attached to a system of levers, worked from a pin rigidly connected with the engine piston rod; so that in each stroke of the engine piston the barrel *O* is moved round a certain distance by the string, and on the return stroke of the engine piston it is returned by the spring to its original position. At the end of one revolution, that is, one forward and one return stroke of the engine piston, the string again pulls the barrel round, and so on, alternately.

The pencil *P* has no horizontal motion; the paper is moved past

it by the movement of the barrel. The amount of traverse in relation to any fixed point bears an exact ratio to the distance of the engine piston from a similarly situated point in its stroke.

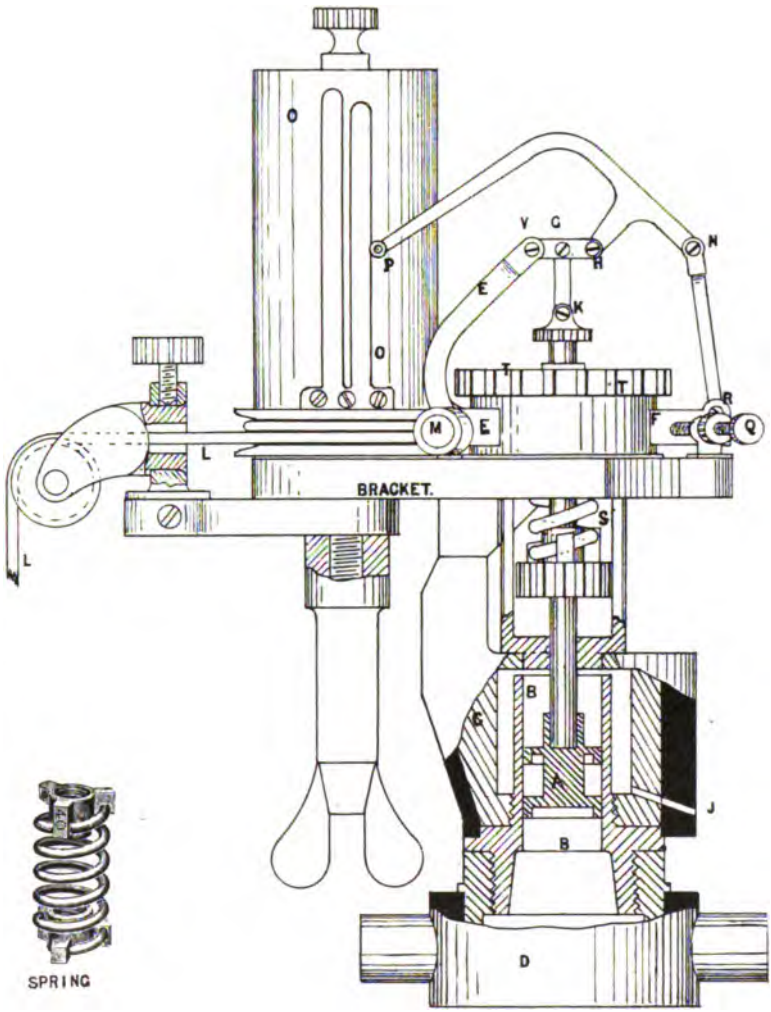


FIG. 244.—M'Innes-Dobbie Indicator.

There is another cylinder, *B* (Fig. 244), in which a piston, *A*, works, with its axis parallel to that of the cylindrical barrel *O*. The upper side of the piston, *A*, is open to the atmosphere through a

small hole, *J*, drilled in the lower part of the cylinder *C*. The lower side of the piston is connected by pipes with the ends of the engine cylinder, and through a cock it can be connected with either end as required. The piston is made of certain area, and loaded by a correspondingly proportioned spring, *S*, so that when the spring is compressed or extended a certain distance, the vertical position of the pencil represents a corresponding pressure *per square inch*.

For various cylinder pressures per square inch various springs are used, corresponding to various scales of compression, namely, 160, 120, 100, 80, 56, 40, 32, 24, 16, and 8 lb. per inch as measured, or marked, on the diagram. Thus, if a spring marked 40 lb. be in use, it means that a vertical movement of one inch of the pencil *P*, which is attached to the piston *A*, represents a difference of 40 lb. pressure per square inch on the engine piston.

The pencil *P* is attached, by a parallel motion, *HGK*, and a rod (shown in the figure), to the piston *A*. When the pressure below the indicator piston *A* is greater than that of the atmosphere, which tends to press the piston downwards, the piston rises correspondingly against the pressure exerted by the spring; but if the pressure below the piston is less than atmospheric pressure, the piston falls correspondingly with the lower pressure in the end of the engine cylinder, with which the indicator cylinder is in open connection.

EF is pivoted on the barrel *B*, so that the pencil *P* can be pressed on, or removed from, the paper at pleasure. A thumb-piece is fitted at *M* for this purpose, and by means of a screw, *Q*, and a stop, fitted on the bracket connecting *O* and *B*, the pencil *P* can be adjusted so as to press lightly on the paper.

Care should be taken, when selecting a spring, that the requisite pressure, both highest and lowest, can be registered on the paper.

Fig. 245 shows the exterior of a M'Innes-Dobbie indicator with the spring enclosed in the cylinder. This type is frequently used for ordinary marine and land type engines; but the external spring type, Figs. 244 and 246, is preferable because the spring, on which the correct indication of the pressure in the engine cylinder principally depends, is not subject to large changes of temperature.

Fig. 246 is very similar in general design to Fig. 245, but a continuous roll of paper is contained within the drum *O* and protected from moisture by a cap on top of the drum. A detent gear is provided for stopping the motion of the drum without dis-

connecting the cord. The drum spring is fitted at *W* below the drum, and its tension can be varied at any time.

Theoretically the motion obtained with the present M'Innes-Dobbie indicator is exactly the same as in the M'Innes type (see 2nd edition), and the peculiar shape of the lever *PHN* in the pantograph motion is merely adopted to obtain a more uniform and direct motion from the piston, with less liability to frictional error. This motion multiplies the piston travel six times at the pencil point. In Fig. 244, a recent improvement is shown by which the cylinder

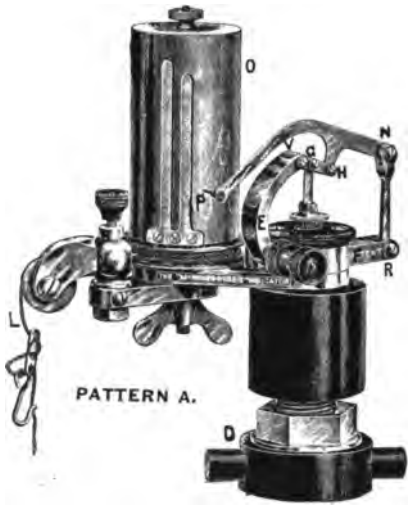


FIG. 245.—Enclosed Spring Indicator.

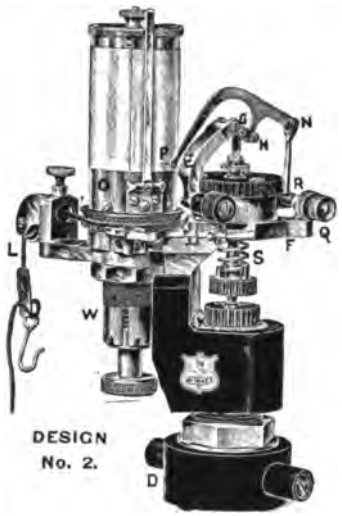


FIG. 246.—External Spring Indicator.

cap can be removed at the same time as the cap *T* and the spring changed more quickly. The piston rod is made of steel tube to secure lightness and strength. The piston *A* is made of case-hardened steel, and the recess shown receives lubrication and grit. These indicators are generally made in three sizes: large, for ordinary steam and ammonia engines; small, for high speed steam engines; and half-size, for motor cars, launches, etc.

The exteriors of the barrel *C*, the cover *T*, the thumb-screw, and the connecting nut *D*, are covered with vulcanised non-conducting material, so that the indicator can be easily handled when warm. To obtain a true diagram, it is necessary to previously thoroughly warm the indicator cylinder and connecting

pipes. A small drain-hole is fitted to the indicator cock, immediately below *D*, by which the condensed water can be blown out, and it is advisable to allow the piston and pencil to work freely for a short time before use, by connecting up each end of the cylinder alternately.

By setting the indicator cock so as to connect the drain-hole with the space below the piston *A*, the atmosphere has free access below the piston, and as it always has free access above *A*, through the holes previously mentioned, the piston can thus be held in a position due to the atmospheric pressure only. The line marked by the pencil *P*, when *A* is so governed, is called the atmospheric line (*GC*, in Figs. 249 and 250). The absolute zero line *OA*, from which the pressures are measured, is drawn horizontally, to scale, 14·7 lb. below this atmospheric line.

The horizontal length of the line made by the pencil may be taken as representing the length of the stroke of the engine piston.

The vertical height of the pencil *P* and the mark made by it at any point above the absolute zero line represents the pressure absolute per square inch on the engine piston at that point, and its position measured horizontally (*i.e.* parallel to the atmospheric and zero lines) represents the relative position of the engine piston in its stroke.

The diagram shows, for one revolution of the engine, the pressure in the cylinder on one side of the piston only. By connecting the indicator with the other end of the cylinder, another figure is obtained, showing the pressure, for one revolution of the engine, on the opposite side of the piston; and from the pair of diagrams the difference of pressure on the piston at any selected point can be measured. For a perfect representation of the pressures in the cylinder at the same instant on opposite sides of the piston two indicators are necessary, but for practical purposes it is quite sufficient to take the diagrams within a few seconds of each other on the same card or paper.

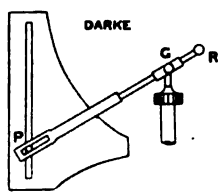


FIG. 247.—Darke Pantograph Motion.

Darke's Indicator (Fig. 247).—The pencil *P* is constrained to move vertically in a slot in a thin plate permanently attached to the crosshead, and the necessary sliding movement is allowed in another slot in the arm (vibrating about the fixed centre *R*) holding the pencil. A sliding motion is

fitted for the arm through a sleeve, *G*, joined to the end of the piston rod.

The paper is contained within the moving barrel, and the outer end of the roll is pulled through a slot and wrapped round the exterior of the barrel. After a diagram is taken, the paper is torn off, and more paper can be drawn from the roll inside and wrapped round the barrel. The barrel can be pawled, by a small ratchet acting in one of a few teeth cut on the lower rim below the paper drum. This arrangement allows the string to remain connected, while preparing to take a new diagram, and, by saving time, allows a number of diagrams to be taken in quick succession.

Crosby Indicator.—The motion of this indicator is shown in Fig. 248, in which the letters correspond to those used in the other figures. All the working parts are made as light as possible, as in Darke's indicator, and in addition the lower part of the spring is made with a ball-and-socket joint, and thus saves some moving weight. The vertical movement of the pencil is about six times that of the indicator piston.

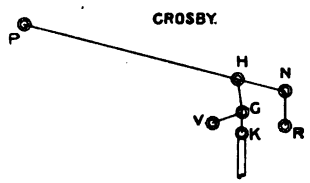


FIG. 248.—Crosby Pantograph Motion.

Wayne Indicator.—This indicator differs from the others described above, as there is no link motion, and the piston is rotary instead of reciprocating. The pencil projects across the end of the rod, rotated by the piston, and marks the pressure on paper, stretched internally on a part of a cylindrical surface, whose axis is parallel to that of the rod and piston. The paper reciprocates, with the engine piston, past the pencil. A spiral spring is fitted outside the indicator piston rod, and resists the rotation of the piston, and as it is outside the cylinder, and therefore not subject to much variation in temperature, the tension exerted by it on the rod and piston is not appreciably varied.

Elementary Diagram.—Fig. 249 shows an elementary diagram which might be obtained from an engine cylinder under instantaneous conditions of admission and cut-off.

GC represents the atmospheric line;
OA „ zero absolute-pressure line, 14·7 lb. below *GC*;
OF „ steam, initial-pressure line, measured vertically;
FE „ steam line, duration of admission;

ED represents the expansion line, after steam is cut off;
DA „ final pressure, after expansion;
BP „ back-pressure line, cylinder open to exhaust during nearly the whole of the stroke.

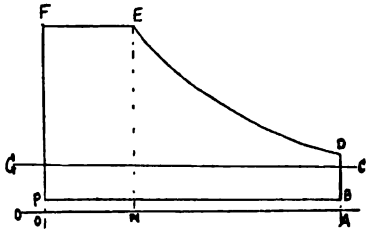


FIG. 249.—Elementary Diagram.

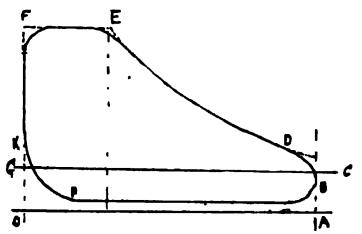


FIG. 250.—Actual Diagram.

Ideal Diagram.—If the expansion is carried to its utmost extent, *B* and *D* will coincide; but this extreme ratio is never reached in practice, because the forward pressure should always be sufficiently in excess of the back pressure to overcome the external resistance of the engine. In some specimens of actual diagrams the points actually coincide, but this is principally from cylinder condensation, producing a lower pressure than would otherwise exist.

Actual Diagram.—Fig. 250 shows a diagram obtained in actual practice, which does not differ much from the elementary diagram shown in Fig. 249. All the points are rounded off. At *F* the full pressure is not obtained until a little after the beginning of the stroke, because the steam is admitted gradually while the valve is uncovering the steam port; for a similar reason, the cut-off is gradual, and the point *E* is rounded off because the pressure falls gradually. The expansion line remains about the same until the point *D*, where exhaust commences, is reached, when the pressure falls gradually until the end of the stroke. The exhaust is not wide open until after the beginning of the return stroke; this produces the rounding of the curve at *B*. The exhaust line is horizontal until *P* is reached, when the exhaust is gradually closed, while the steam in the cylinder is compressed behind the piston and the pressure rises until, just before the end of the return stroke, admission commences, when the pressure rises still further, and frequently attains the initial pressure ready for the next forward stroke.

The rounding at all these points affects the working of the

engine, generally in a favourable manner. Thus, at *D* there is no sudden fall of pressure when the piston is changing its direction of motion. From *P* to *K* the pressure rises gradually, and not suddenly, the piston is brought slowly to rest without shock, and the full pressure is reached at about the time it is required for the forward stroke. This rounding at *PK* is called the *cushion or compression corner.* The steam and back pressure lines should generally be horizontal, and the admission and final pressure lines should generally be vertical; but in practice, as will be shown later, the figures vary considerably, and each difference is due either to some peculiarity of the engine or its working. These are not necessarily either faults of construction or of working, and are frequently the outcome of economical experience.

Errors due to the Indicator.—Unless the indicator is kept in good order and perfectly clean, the piston, pencil, and other sliding parts have a great tendency to produce jerky movements. *Friction* also reduces the mean pressure recorded, by raising the back pressure, and by decreasing the height of the forward pressure.

The variations in the pair of diagrams shown in Fig. 251 have been produced by friction, probably between the indicator

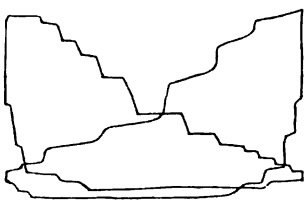


FIG. 251.—Friction of the Indicator.

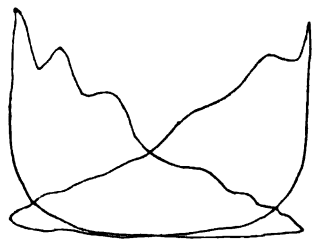


FIG. 252.—Undulation of the Indicator.

piston and its cylinder, from gritty matter or lack of proper lubrication. Instead of a gradual curve there are a series of steps, formed by undue frictional resistance holding the piston until the difference of pressure overcomes it and produces some small movement. The example shown is a very bad case; in practice it is generally less accentuated, and more nearly approaches that due to the undulation of the indicator.

Undulation or vibration of the indicator is shown in Fig. 252, and is produced from various causes—either due to the indicator not being suitable to the number of revolutions of the engine, or to the

spring not being suitable to the pressure. The example was taken with an old pattern Richard's indicator, and that shown in Fig. 263 was taken from the same engine, at about the same power and number of revolutions, with a later pattern indicator. The substitution of a stronger spring sometimes overcomes the effect of undulation by allowing less travel of the indicator piston; and by shortening the travel of the paper past the pencil, horizontally, some improvement can also be made. The working length of one of the levers which move the string can generally be altered by removing one of the pins in the lever connections and fitting it into another hole provided for the purpose.

If the indicator piston is moved at great velocity by a very quick reciprocation of the engine, and the weight of the moving parts of the indicator is unnecessarily great, it increases the tendency which the pencil always possesses to rise too high and fall too low. The frictional resistance of the piston in the cylinder, the pencil on the paper, and in the joints of the mechanism, tend to check this excessive movement of the pencil; but when the pencil arrives at the upper and lower extremities of its travel, the frictional resistance tends to keep the pencil either above its normal position when falling, or below its normal position when rising, and therefore the area of the figure contained within its boundary lines may be slightly larger than the perfect figure, obtainable if these defects did not exist. Other errors possibly occur from inaccurate fitting of the joints in the mechanism—inaccurate calibration of the spring, when equal degrees of compression are not exactly to scale; and variation in the length of the string (or the virtual length, if pin or sliding joints are used in connection with it), caused by difference in the tension exerted by the spring.

All these errors are nearly avoidable, and, although they should never be overlooked, can generally be neglected for the practical purposes for which the indicator is used.

The principal error affecting diagrams in actual practice is caused by wiredrawing in the pipes connecting the indicator cylinder with the engine cylinder, and for naval purposes the pipes have recently been increased in area to obtain more accurate reading. The velocity of the steam through these pipes should not as a rule exceed 6000 feet per minute, from which the area can be calculated.

Faults shown by the Diagram.—There are in practice many

faults which may be gathered from a study of the diagrams taken from a cylinder, as well as the useful information relating to the horse-power, steam consumption, and other matters of calculation. The faults are given in detail as follows:—

Length of the String.—If the string be not the correct length, the barrel of the indicator comes to rest before the end of the piston stroke, and therefore it is necessary to test this adjustment by taking a trial diagram. The effect is shown by the general appearance of the admission and final pressure lines, one of which will probably be vertical, with square corners both at top and bottom.

Wiredrawing.—If there is insufficient opening of either the steam or the exhaust ports, wiredrawing occurs, but care should be taken not to confound this result with that obtained by using an early cut-off when "linked up."

If the opening to steam is very much restricted, the steam pressure is reduced before the point of cut-off is reached, producing a gradual fall of pressure on the admission line before expansion commences. This is shown in Fig. 253, where the highest pressure is not reached until a little after the beginning of the stroke, and at the point of cut-off it has fallen to *L* instead of remaining at *E*. The absolute pressure at this point is represented by the height *NL* instead of *NE*.

Restricting the opening to exhaust is a very serious loss, and the effect is shown in Fig. 254. The consequent increase of back

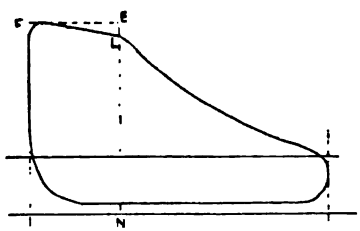


FIG. 253.—Steam wiredrawn.

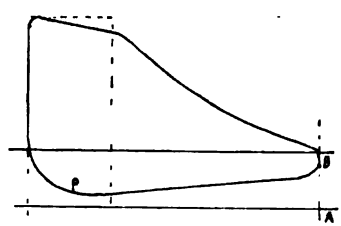


FIG. 254.—Steam and Exhaust wiredrawn.

pressure reduces the mean effective pressure on the piston, and thus reduces the power indicated by the engine. The defect is a common one on early trial trips, but is generally remedied after discovery by enlarging the exhaust ports both in the slide valve and in the cylinder false face. To obtain an unrestricted exhaust, negative exhaust lap is frequently used in fast-running engines.

In Fig. 254 the result of wiredrawing both the steam and exhaust is shown. The exhaust pressure is not fully released at the beginning of the return stroke, as shown by *BP*, which is not horizontal.

Leaky Slide Valve.—If the slide valve is leaky, the steam continues to enter the cylinder after cut-off has taken place, and instead of the expansion line being as shown in dotted lines in Fig. 255, there is a continuing higher pressure, as shown by the full line *EQB*.

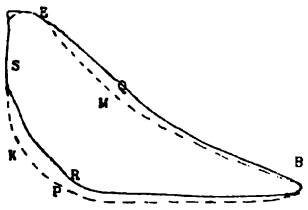


FIG. 255.—Leaky Slide Valve.

Some steam leaks past the steam edge of the valve, without entering the cylinder, into the exhaust side, and raises the back pressure, as shown by *BRS* instead of *BPK*, thus reducing the power obtained in the cylinder with a greater expenditure of steam. The loss with fast-running engines is considerably less than with slow-running engines, where the steam has plenty of time to leak past the edges of the valve. In stage-expansion engines a small percentage of leakage is not so important in the high and middle pressure cylinders, because the leakage is eventually used in the low-pressure cylinder, and not entirely wasted. The flat valve is supposed to be less liable to leakage than the cylindrical slide valve, and therefore flat valves are frequently fitted to low-pressure cylinders. It is especially important that no leakage should occur through the low-pressure slide valve, because, in addition to the actual leakage and consequent waste of steam, there is a tendency to increase the back pressure by decreasing the vacuum in the condenser (as shown in the figure, which is somewhat exaggerated).

Note.—In all diagrams the front end of the cylinder is next the crank.

Leaky Piston.—A pair of diagrams are shown in Fig. 256 which were taken from the high-pressure cylinder of a compound engine. The right-hand diagram, taken from the back end of the cylinder, varies but little from that obtained in ordinary practice, when properly "linked up" for a low power.

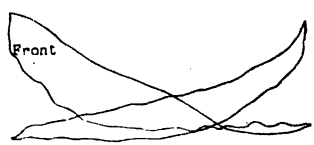


FIG. 256.—Leaky Piston Packing.

The diagram taken from the front end is abnormal, showing large fluctuation of back pressure, with an average above the

normal. Pressure has apparently been allowed to pass from back to front, but not from front to back, as shown by the front-end diagram; leakage is also evident from back to front, as shown by the back-end diagram. The back-end diagram shows an almost immediate fall of pressure at the beginning of the stroke, and there is a corresponding rise on the opposite side of the piston. This points to leakage from back to front. The front-end diagram shows a normal fall of pressure from the beginning to the end of the stroke; but on the return stroke the exhaust or back pressure is too high, and confirms the leakage from back to front. Such leakage could be caused by either the packing ring not fitting properly between the junk ring and the body of the piston, or by the tongue piece not properly covering the opening between the ends of the packing ring. The piston, when pushing the packing ring and tongue piece before it, might be expected to maintain a steam-tight joint between it and them; but when the piston is dragging the ring and tongue, some leakage might be expected either by way of the tongue or through the space left between the junk ring and the body of the piston, and thence through the space between the packing ring and the flange. In this instance the packing ring was found a little slack between the junk ring and the flange of the piston, and the tongue was in a similar condition. A thin liner was removed from between the junk ring and the piston, no alteration was found necessary in the tongue, and subsequent diagrams were of a normal character.

Such a leak is difficult to correct, as it is always possible to nip the packing rings too closely. Accurate fitting can only be obtained by removing the piston from the cylinder, which may take a long time, as frequently the piston cannot be withdrawn from the rod without considerable trouble and delay. When the piston is replaced on the rod plenty of dry black-lead should be smeared on the joining surfaces, and little trouble is then experienced in subsequent removal.

Cylinder worn unevenly.—In Fig. 257 some diagrams, taken some years ago from a L.P. cylinder in H.M.S. *Colossus*, are shown. The cylinder barrel was worn in such a way that there was considerable variation of the actual diameter in which the piston worked. The packing ring therefore continually varied the opening at the tongue piece or sliding joint, and allowed varying leakage through this opening. The diagrams were taken at very low pressure and power, the initial pressure in the cylinder being

only 5 to 6 lb. absolute, with a vacuum of about 28 inches in the condenser. The points to notice are: that up to the point of cut-off the forward pressure is nearly maintained, because the leakage is not great enough to appreciably lower the pressure; towards the middle of the stroke, after admission ends, the pressures on the opposite sides of the piston rapidly equalise, and for the latter part of the stroke there is no difference of pressure, because all the steam has taken a short cut to the condenser; the opening to the condenser is open nearly the whole of the stroke, and consequently the exhaust line is almost horizontal throughout the stroke; the cushioning corners are peculiar—at one end no compression is apparent, and is probably prevented by leakage, while at the other end some cushioning is apparent, indicating less leakage, and that the working barrel is less worn at this part.

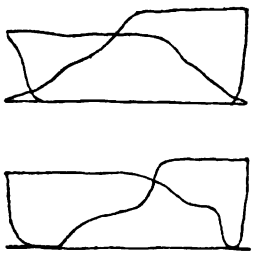


FIG. 257.—Cylinder worn.

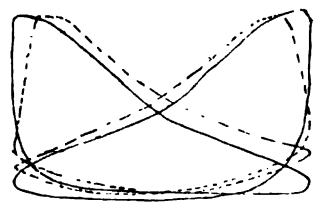


FIG. 258.—Angular Advance too small.

Angular Advance of the Eccentric too small.—In this case all the operations of the valve, including admission, cut-off, release, and compression, are late; and instead of the diagram shown in full lines in Fig. 258, one similar to that shown in dotted lines results. This diagram shows similar characteristics for both the forward and the return strokes; whilst an alteration in the length of the rod produces characteristics of opposite kind for forward and return strokes, as exemplified in Figs. 260 and 262.

Angular Advance of the Eccentric too great.—In this case all the operations of the valve are too early, and a diagram such as shown in Fig. 260 results, with perhaps steam and exhaust loops. The characteristics obtained by advancing the eccentrics are similar for both the forward and return strokes.

The effect of *linking up*, or shortening the link, produces a similar result, by reducing the travel of the slide valve on the false face.

After an engine has been designed and completed, the angular

advance is sometimes altered to produce a more accurate result in the power developed, or to suit a difference in the pitch of the screw. If the angular advance is decreased, a later cut-off is obtained, and a greater power can be obtained from the engine, while an increase produces an earlier cut-off, and allows less power to be developed. It does not follow that either operation will produce a difference in the economy of steam, but within moderate limits, as pointed out elsewhere, an early cut-off produces economy.

Slide Valve incorrectly set on the Rod.—In this case the slide valve has too much *steam lap* at one end and not sufficient at the other, while the exhaust lap is insufficient at the first end and too great at the latter. Diagrams taken in actual practice are shown in Figs 259 to 263.

In Fig. 259 the rod is much too long. There is no semblance of equality in either initial or mean pressures, and the steam admitted at the back end is considerably over-expanded; while probably an exhaust loop would have been formed, if the heat of the cylinder walls had not partially re-evaporated the condensation water, and thus produced steam of a slightly higher pressure than the back pressure on the piston during the return stroke. Although steam is cut off much later at the front end, there is a rapid diminution of pressure, because the heat, abstracted during the previous stroke, has to be replaced in the cylinder walls. The mean cut-off, as nearly as could be measured, was at about 30 per cent of the stroke, and the diagram was taken at a comparatively low power.

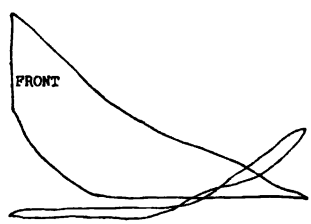


FIG. 259.—Rod much too long.

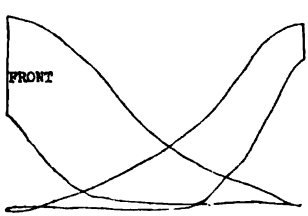


FIG. 260.—Rod too long (moderate power).

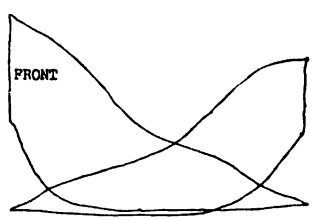


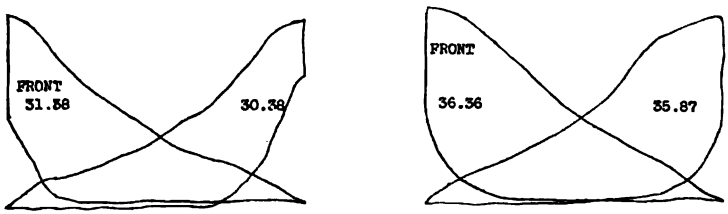
FIG. 261.—Rod too long (high power).

In Fig. 260 the rod has been shortened, but is still too long. This is shown by the inequality of the mean and initial pressures,

the cut-off being too early at the back end and too late at the front end. The too early cut-off at the back produces over-expansion, which is shown by an exhaust loop.

Fig. 261 shows a pair of diagrams taken at the same slide-valve setting and cut-off, about 33 per cent, but at a higher power and speed. The same characteristic features are shown as in Fig. 260, but there is no exhaust loop; this may be due to the slight difference in the cut-off, or to less cylinder condensation, with a higher rate of revolution and a higher power.

Figs. 262 and 263 show diagrams taken after shortening the rod still more, and the mean and initial pressures are now nearly



FIGS. 262, 263.—Rod about Correct Length.

equal. The exhaust loop is not evident, and for ordinary working, with the link considerably shortened to cut off at 30 per cent, instead of 60 per cent of the stroke when working at full power, the diagrams are satisfactory.

Clearance.—The effect of clearance is to vary the mean pressure obtained in the cylinder. The piston cannot be allowed to touch the cylinder cover or end at each end of the stroke, and consequently a small amount of space is allowed for safety. When the engine bearings wear, and so alter the exact position of the piston in the cylinder at the ends of the stroke, it tends to alter the virtual length of the rods, generally shortening them, and thus brings the piston downwards in a vertical engine. The steam passages between the slide valve and cylinder must also be considered. The sum of these several clearance volumes is ascertained, and an equivalent calculated in a fraction of the length of the stroke. The cut-off takes place also at a fraction of the stroke. The sum of these fractions, *less* the fractional amount of steam left in the cylinder at the end of the stroke, actually represents the volume of steam admitted at each stroke. The effect of clearance is shown graphically in Fig. 264, where the expansion

line *ED* is somewhat higher than *EB* in the elementary diagram in which clearance has not been considered. Clearance affects the rate of expansion, for, in addition to the initial volume *FE* of steam admitted to the cylinder between the beginning of the stroke and the cut-off, an extra amount *FF'* is admitted to fill the clearance at each stroke.

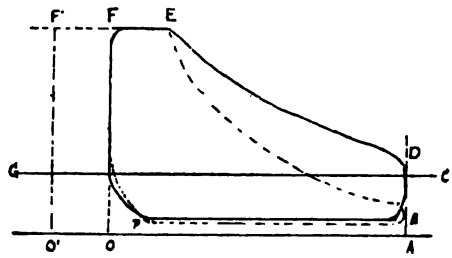


FIG. 264.—Effect of Clearance.

The actual amount is best obtained from the calculation of the weight of steam used per stroke, which is explained later. For the present calculation, the amount of steam *left* in the cylinder at the end of each stroke is neglected.

Let $\frac{1}{r}$ represent *FE*, the fraction of the stroke at which steam is cut off; and let $\frac{1}{c}$ represent *FF'*, the clearance, as a fraction of the stroke-volume.

$$\left.\begin{matrix}\text{Then the volume of steam} \\ \text{admitted at each stroke}\end{matrix}\right\} = \frac{1}{r} + \frac{1}{c} = \frac{r+c}{rc}.$$

At each stroke this volume of steam is expanded into the volume of the cylinder struck out by the piston, added to the clearance, that is, into a volume represented by $1 + \frac{1}{c}$.

$$\text{Therefore the } \textit{actual} \text{ ratio of expansion} = \frac{1+\frac{1}{c}}{\frac{r+c}{rc}} = r\left(\frac{1+c}{r+c}\right).$$

For example, take a cylinder in which the cut-off, as shown by the slide valve (or *apparent* cut-off), is at one-fourth stroke, and suppose the clearance is equal to one-eighth the stroke-volume of the cylinder—

$$\text{Then the } \textit{actual} \text{ ratio of expansion} = 4 \times \frac{1+8}{4+8} = 4 \times \tfrac{9}{12} = 3.$$

So that instead of obtaining *four* as a ratio of expansion, the actual ratio is reduced to *three*, from the effect of clearance.

The waste of steam resulting from clearance is reduced by the compression at the end of the return stroke. When the pressure is raised by compression in the clearance spaces, just before the point of admission, to the initial pressure, the loss from clearance is very small; but as this increases the mean back pressure, there is a decreased effective mean pressure. In simple or single-stage expansion engines it is not economical to compress the steam to this extent; but for higher stage expansion engines, when running at low powers, the economy of steam can be greatly improved by "linking-up," until the pressure of the cushioning steam in the H.P. cylinder is nearly equal to the initial pressure, thus producing the highest practicable total rate of expansion. The separate link is very useful for this purpose.

Steam Loop.—When the pressure of the cushioning steam exceeds that of the initial steam, it shows itself by a loop, as seen in Fig. 265. It should not in practice be allowed to quite attain this pressure, but something just below it. In calculating a diagram with a loop, the area of the loop should be deducted from the area of the diagram, because it represents work done against the direction of motion of the piston, and thus reduces the *mechanical efficiency* of the engine.

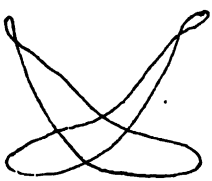


FIG. 265.—Steam Loop.

Exhaust Loop.—On reference to Fig. 260, a small loop is seen in the left-hand corner, at the tail end of the diagram, taken from the back end of the cylinder. This is due to the pressure in the cylinder falling below that of the ordinary expansion curve, which is nearly adiabatic, and is the result of either—

1. Too high initial pressure, combined with too early cut-off, generally obtained by linking up too much; or
2. Steam jacketing being insufficient to prevent undue cylinder condensation.
3. Incorrect setting of the slide valve.

Exhaust loop seldom occurs in the L.P. cylinder, because the range of temperature in the cylinder itself is frequently less than in the other cylinders, and the cut-off is possibly later; both of which are reasons for less apparent condensation in the L.P. cylinder.

In the H.P. cylinder the initial pressure may be unnecessarily great for the power developed by the engine. For naval service it is usual to keep an otherwise unnecessarily high pressure for possible manœuvring purposes, and for Belleville boilers a minimum pressure of 140 lb. is recommended. A moderately high pressure is sometimes more economical for auxiliary purposes.

The economical advantages of high expansion cease when exhaust loop occurs, so far as the cylinder itself is concerned, but it is still admissible if it does not affect the smooth working of the engine, and if the power developed in the cylinder is not below that developed in another cylinder of the same engine. In some instances a judicious use of the cylinder jacket may cause the loop to disappear.

Measurement of the Weight of Steam used, from the indicator diagram.—In Fig. 266, *OA* represents the zero line of absolute pressure; *O'A*, the length of the diagram, represents the stroke-volume; and *OO'* represents the clearance on the same scale. Take a point, *P*, in the expansion curve after cut-off has taken place and where the steam contains least moisture, and drop a perpendicular, *PM*, to *OA*. Then, (*OM* × area of piston) represents the volume of steam apparently admitted to the cylinder during the stroke, and *PM* represents the absolute pressure. The weight of a cubic foot of steam at a pressure, *PM*, can be found in the table at the end of Chapter II., and the weight of steam apparently admitted at each stroke can then be calculated.

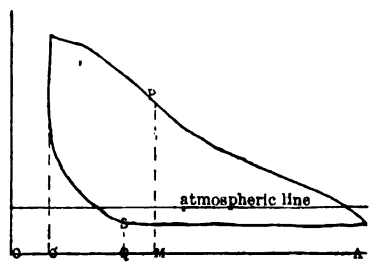


FIG. 266.—Weight of Steam used.

Next, take a point, *S*, in the exhaust curve just after cushioning or compression has commenced, and drop a perpendicular, *SQ*, on *OA*; then, (*OQ* × area of piston) represents the volume of steam which does not escape from the cylinder at each stroke, and *SQ* represents its absolute pressure. The weight of a cubic foot of steam at a pressure, *SQ*, can be found on page 18, and the weight of steam which does not escape at each stroke can be calculated.

If it be assumed that the steam, at the points of measurement, is dry saturated, then the *difference* between the weight of steam apparently admitted during the stroke and the weight of steam

which does not escape is the actual weight of steam which is used at each stroke. If w represents this amount, then the weight of steam used per I.H.P. per hour is equal to $w \times$ number of strokes per hour divided by the I.H.P.

In practice this amount is always below the actual amount, and it can never be in excess unless the steam is superheated considerably, and even then the result is not usually great enough, because the steam generally contains moisture which is not perceptible on the diagram.

Combining Diagrams of Stage Expansion Engines.—The diagrams taken from the various cylinders of a stage expansion engine are generally on different scales, such as are suited to the pressures used in them. In combining them, therefore, the scale must be corrected for each when plotting them on the same curve or figure. The horizontal length of each figure must be proportioned to the cylinder volume, and the vertical height which represents the pressure must be proportioned to a certain scale. If there is more than one cylinder in one stage of the expansion, the volumes must be added together to represent the volume of the virtual single cylinder on the combined figure.

After the diagrams are obtained either by actual practice or by precise calculation, the procedure is graphical, and as follows:—

1. *When Clearance is not taken into Account.*—A base

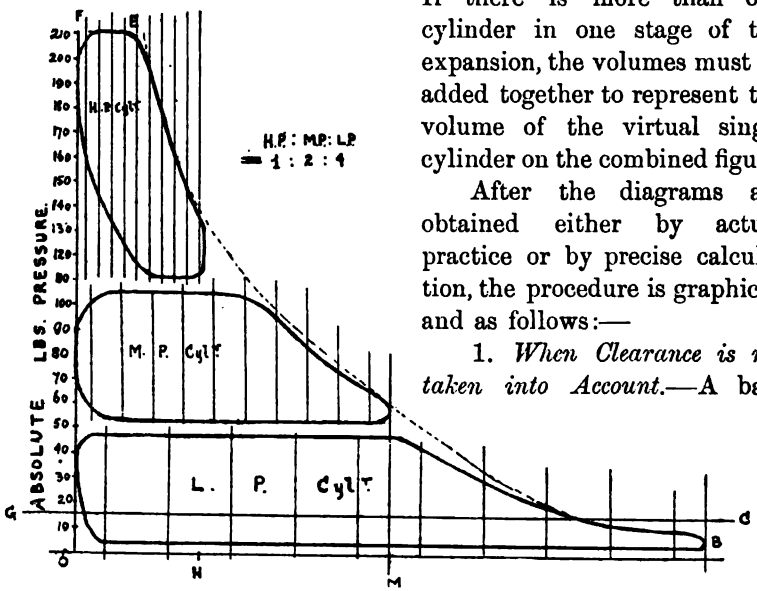


FIG. 267.—Combined Diagrams, neglecting Clearance.

line, OB (Fig. 267), is taken to represent the volume of the L.P. cylinder or cylinders; then mark off OM to represent to the same

scale, the volume of the M.P. cylinder, and *OH* to represent the volume of the H.P. cylinder. Draw a perpendicular, *OF*, representing the absolute pressure of the steam to scale, so many pounds to one inch. Then construct the figures from the diagrams, taking care that the length and height are on the new scale. The figures obtained will be something like those shown. This combined figure is very similar to what might be expected if the steam were cut off at *E*, and the expansion carried out in a single cylinder.

2. *When Clearance is taken into Account.*—The actual amount of steam which is expanded in the H.P. cylinder, and so on through the various stages, is increased by the clearance spaces which must necessarily exist. Therefore, in setting out the various cylinder volumes, the actual volume of each cylinder *plus* the clearance is more nearly correct. This is shown in Fig. 268. Each cylinder volume is credited with the addition due to its clearance volume.

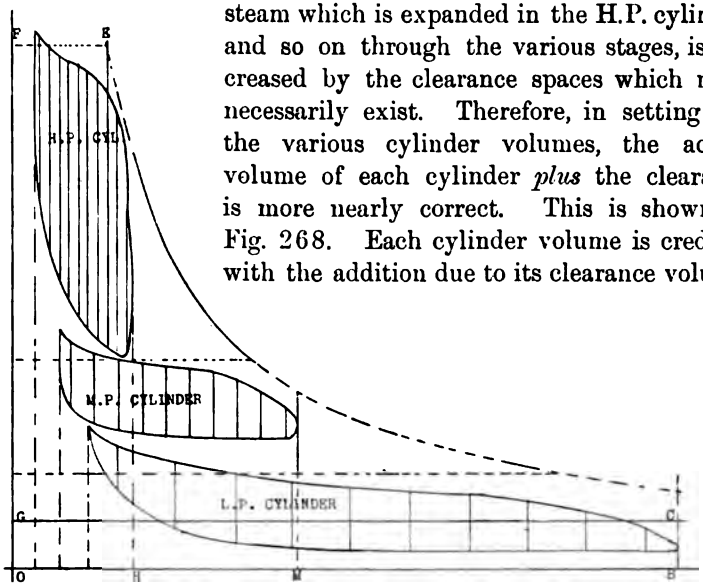


FIG. 268.—Combined Diagrams, Clearance considered.

3. *When Clearance and the Amount of Steam left in the Cylinder at the End of each Stroke are taken into Account.*—For a more precise method, the amount of steam remaining in the cylinder after compression begins must be deducted from the amount of steam in the cylinder at the instant of cut-off, and therefore a deduction should be made from the clearance steam for each cylinder, and it can be calculated by the method explained for the measurement of weight of steam used (Fig. 266). If the engine is "linked-up" considerably, the clearance volume is frequently reduced by one-half, and therefore in Fig. 268 the

clearance spaces allowed must be reduced to the same extent. The conclusion is, therefore, that the combined diagram is somewhat between the extreme limits shown in Figs. 267 and 268, and the expansion curve, nearly hyperbolic, lies somewhere between the two curves shown.

Receiver Drop.—From all three methods, it is evident that there is an apparent loss of work by using a three-stage expansion engine instead of expanding in a single cylinder, of the same volume as the L.P. cylinder. If the steam could be maintained in a state of dry saturation, and there were no cylinder condensation, the loss by using stage expansion would be very great; but it is found in practice, as explained elsewhere, that the great range in temperature obtained by expanding in a single stage produces excessive condensation loss, and thus the actual economy is greater with the stage expansion engine. In practice, if steam of similar initial pressure is expanded in a single stage instead of three stages, as shown in the figures, the forward pressure during nearly the whole of the stroke is lowered by warming the cylinder; and during the latter part of the stroke, corresponding to the L.P. cylinder in a three-stage engine, is but slightly greater than the back pressure, and in some cases actually below it.

Cippolini Continuous Double-diagram Indicator.—For close and accurate testing of reciprocating steam engines it is convenient to use a continuous automatic arrangement for taking the diagrams. In Fig. 269 a type of M'Innes-Dobbie external spring indicator is shown, adapted to the purpose of taking complete diagrams simultaneously and continuously from both ends of the engine cylinder. The string is attached to the engine piston rod in the usual way, and by an automatic arrangement records every revolution of the engine on a paper cylinder, *C*, which is also in connection with an electrical timing arrangement, and thus the rate of revolution is also automatically recorded. The indicator can be set so as to record a reading automatically every 25, 50, or 100 revolutions, or as required. The paper is contained within the barrel, and is automatically unwound from the drum at each period of registration.

The advantages of an automatic apparatus of this type, which avoids all personal error, are obvious. Each pair of diagrams can be taken either together or separately and entirely independent of the confusing lines of the usual continuous diagram. A full

description is generally supplied with the apparatus, and it is not necessary to enter here on a more detailed description.

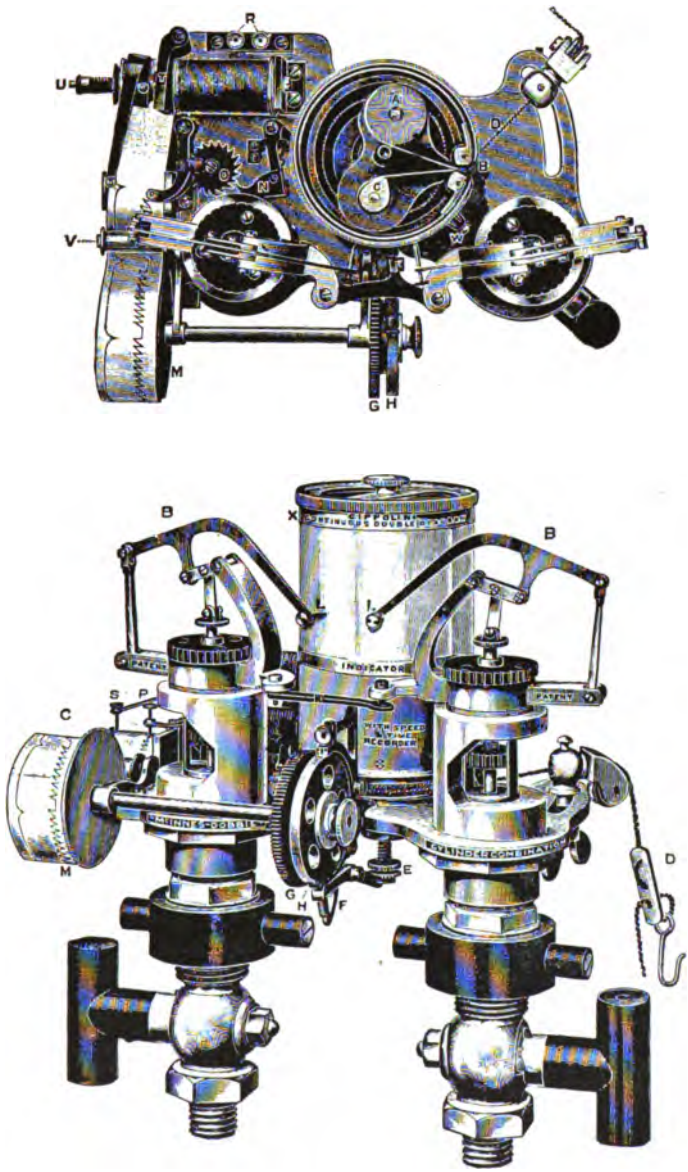


FIG. 269.—Cippolini Continuous Indicator.

CHAPTER XXII

EFFICIENCY OF THE STEAM AND MECHANISM

Introductory.—In Chapter IV. a general idea of the laws of expansion under various assumed conditions has been given, and there now remain to be considered the variations produced in actual working, with their effects on the efficiency of the steam and their bearing on the efficiency of the engine.

The *mechanical efficiency* of the engine is almost entirely independent of the steam, and is governed particularly by the frictional resistances of the mechanism. These may or may not be affected by the increased temperatures and differences of temperature of the working parts and their proper lubrication, but under good conditions the difference in mechanical efficiency should be only very slight. For the present only the efficiency of the steam, which is the ratio of the useful work done by it in the engine to the total heat (in foot-lb. or b.t.u., according as the useful work is measured in the same units), conveyed to the engine from the boiler, is under discussion.

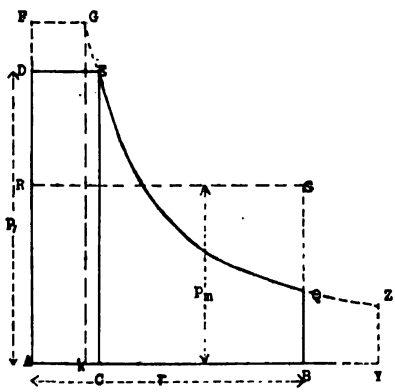


FIG. 270.—Influence of Boiler Pressure.

Influence of Boiler Pressure.—For convenience of demonstration of principle, the hyperbolic or other law of expansion may be assumed, as in Fig. 270. In this the

work done during admission = *ADEC*, and
work done during expansion = *CEQB*.

By increasing the ratio of expansion from $\frac{AB}{AC}$ to $\frac{AY}{AC}$, the quantity of work done by the steam is increased, without admitting any more steam into the cylinder, and its efficiency is increased by using increased expansion.

By increasing the boiler, or admission, pressure from AD to AF, and cutting off admission at k, instead of at C, the work done during admission = $AFGk$, and the work done during expansion = $kGQB$. The area $AFGk = ADEC$, assuming hyperbolic expansion, and therefore the work done during admission is the same as before; but the ratio of expansion is increased, because steam is now cut off earlier in the stroke of the piston. More work is done by the steam, as represented by the area $DFGE$, and greater economy is obtained in its use and therefore greater efficiency, because only about the same quantity of heat is necessary to produce the higher initial pressure.

The increase in initial pressure does not increase the size of cylinder necessary for the same ratio of expansion, although more work is obtained. An increase in ratio of expansion, without increasing the initial pressure, increases the work done, but requires a larger cylinder in which to carry out the expansion; in other words, an increase in size of cylinder is necessary to obtain greater economy and efficiency, unless it is accompanied by an increase of pressure.

Effect of "Linking-up."—By linking-up, the admission of steam to the cylinder is cut off earlier, and therefore the ratio of expansion is greater. Increased work is obtained during expansion, and also greater economy and efficiency of the steam.

When the engines are eased down, the pressure at admission is reduced by *throttling* the steam at the regulating valve. The final pressure, being almost constant at all powers, is not reduced in the same proportion as the admission pressure; and the fall in pressure being thus limited makes an actual decrease in the ratio of expansion, and thus for some part of the latter part of the stroke no work is done, because the pressure on the two sides of the piston is the same. By *linking-up*, while the same work is done by the steam in the cylinder, the initial pressure is increased and the cut-off is earlier; and greater work being done during expansion, a greater economy and efficiency are obtained from the steam.

Cylinder Condensation.—At the beginning of the stroke of the piston after exhaust has taken place, steam is entering the cylinder. The cylinder and all parts of the metal which have

been in contact with the exhaust have been cooled by its lower temperature, and consequently the entering steam does not act with its full energy, part of which is absorbed in reheating the cylinder metal. Condensation takes place, reducing the pressure, and the steam becomes moist; the moisture is deposited on the walls of the cylinder, and the deposit increases the rate of conduction of heat from the steam to the walls, and thus tends to increase the loss.

There is also a slight condensation due to the conversion of some of the heat energy in the steam into work on the pistons; but this liquefaction is partly counteracted by the re-evaporation during expansion, because the expanded steam requires less total heat to keep it in a state of saturation or dryness.

Complete and Incomplete Expansion.—For so long as the forward pressure (*OY* in Fig. 271) on one side of the piston is

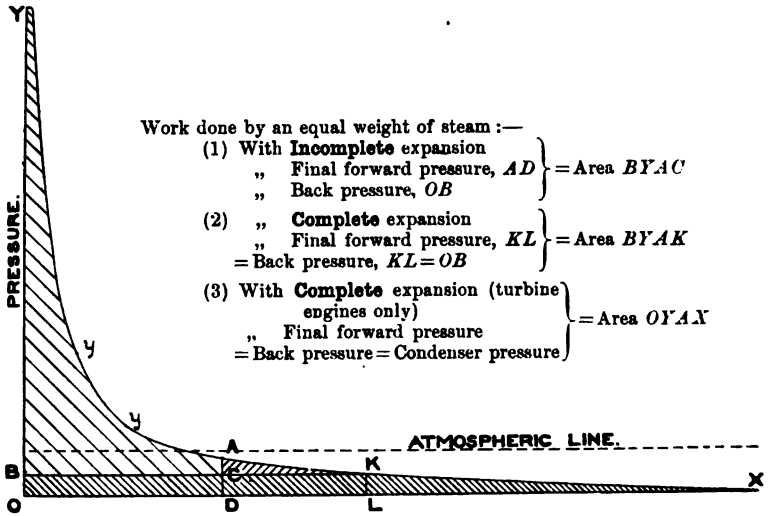


FIG. 271.—Complete and Incomplete Expansion. Work done by 1 lb. weight of Steam.

greater than the exhaust pressure *KL* on the opposite side, it is evident that more work can be obtained by further expansion. By increasing the expansion until the final pressure *KL* becomes almost equal to the exhaust pressure, the maximum work is obtained from the steam and the expansion is then complete. During the latter part of the stroke of the piston, however, very little work is added to that already done in the earlier part: because the difference of pressure on the two sides of the piston is so small,

and may be insufficient to overcome the frictional resistance of the mechanism and, therefore, be powerless to produce motion. In practice, expansion is seldom continued below 7 lb. absolute pressure, and is never entirely complete for the above reasons and also because the increased size of cylinder necessary for complete expansion is not actually economical in space and weight, as well as adding to the cylinder condensation losses. As can be seen from the indicator diagrams in the previous chapter, there is generally a rapid fall of pressure (to the exhaust pressure) towards the end of the stroke of the piston.

In the case of turbines, the expansion may be continued without inconvenience down to the absolute pressure of the condenser, and the increased work obtainable from the same steam is represented in the figure by the area $OBKX$, where OX represents the absolute pressure in the condenser.

In the figure:—

the area $BYAC$ = maximum work obtainable with incomplete expansion, final pressure AD, and back pressure $CD = KL$;

the area BYK = maximum work obtainable with complete expansion;

the area OYX = maximum work obtainable with turbines, back pressure line being OX.

Steam Jacket.—To counteract the losses from cylinder condensation the steam jacket was introduced, and still exists in nearly all large engines. The steam is admitted into a space between the liner and the cylinder, and any water produced by the transfer of heat to the steam in the cylinder is drained away as required.

Under certain conditions of working, the steam jacket produces great economy by keeping the steam in the cylinder dry, and thus performing efficient work through its medium. The steam jacket, however, must be given sufficient time to transfer its heat to the steam in the cylinder; with a high rate of revolution particularly, and with great piston speed indirectly, it ceases to be efficient, and may result in an actual loss of heat. In the Navy the steam jacket is seldom a source of economy when used for over one-fourth power, but no distinct line can be drawn for all engines; it is a case of each engine standing on its own merits.

The top and bottom ends of the cylinder are theoretically useful

for steam jacketing, but in practice they are generally inoperative; this is due to the slag or core surface of the metal and to deposits of grease or oil, which prevent any rapid transference of heat. Thus jacketing the ends of the cylinders is of little use in producing economy, and at the same time heat is radiated from them into the engine room, producing unnecessarily high temperatures without any compensating useful effect.

The interior surface of the wall of the cylinder (*i.e.* the liner barrel) is kept in a clean and polished state by the reciprocation of the piston, and acts quickly as a conductor of heat, but it is obvious that the jacket heats the exhaust steam also. The quantity of heat transferred to the exhaust is generally very small, and not of very great consequence, due to the fact that there is little moisture present, and that the rate of conduction is very slow. Any heat imparted to the exhaust is a loss in a single-stage expansion engine, because the heat passes into the condenser without doing any work. It is probably for this reason that there is seldom any practical economy in jacketing the L.P. cylinder of a compound engine, although under the same conditions there may be a fair economy by jacketing the H.P. and M.P. cylinders.

Thermo-dynamically the L.P. jacket is the most efficient, and this would be the case practically if no re-evaporation occurred at the end of each stroke. Unfortunately, however, it is almost impossible to efficiently jacket this cylinder, so that steam is formed by re-evaporation, and passes directly into the condenser instead of doing work in another cylinder.

If economy is to be gained by jacketing, the pressure in any one jacket must be at least as great as the initial pressure in that cylinder, or the jacket steam will extract heat instead of imparting it. This is arranged for in practice by fitting gauges in connection with the jacket and the receiver, side by side for each cylinder, so that they can be readily seen and regulated.

Jackets are most useful for warming the engines in the first instance, and for keeping them warm when waiting orders; they appear to be particularly necessary for naval engines, except those of the smallest size, where small weight is of greater importance.

The following table shows the result of jacketing the cylinders of H.M.S. *Argonaut* and *Diana*, and it should be noticed that it bears out the foregoing remarks, which were originally made before these trials had been carried out:—

	I.H.P. Developed.	Fraction of Maximum Power.	Cylinder Jackets in Use.	Pressure of Steam.		Cut-off in H.P. Cylinder.	Total Expansion. (Neglecting Clearance.)	Steam Consumption in lb. of Water used per I.H.P. per hour.		
				Boilers.	H.P. Receiver.			Main Engines.	Auxiliary.	Total.
						Per cent.				
H.M.S. *Argonaut.*	18,000	1	All	299	237	71	8·5	16·64	1·88	18·52
			L.P. only	290	242	71	8·5	16·15	1·88	18·03
			None	292	236	71	8·5	15·75	1·88	17·63
	13,500	$\frac{3}{4}$	All	280	244	43	12·5	15·85	1·59	17·44
			M.P. and L.P.	280	234	53	10·4	15·87	1·59	17·46
			None	251	230	53	10·4	15·44	1·59	17·03
	3600	$\frac{1}{5}$	All	285	164	27	15·7	16·32	3·57	19·89
			M.P. and L.P.	294	166	27	15·7	16·34	3·57	19·91
			None	275	168	27	15·7	16·26	3·57	19·83
H.M.S. *Diana.*	8000	$\frac{5}{6}$	All	151	145	65	6·4	17·75	1·93	19·68
			L.P. only	149	148	65	6·4	17·94	1·96	19·90
			None	151	144	65	6·4	17·57	1·93	19·50
	6400	$\frac{2}{3}$	All	146	121	60	6·8	18·06	2·19	20·25
			M.P. and L.P.	136	119	60	6·8	17·91	2·65	20·56
			L.P.	136	117	60	6·8	17·84	2·66	20·50
			None	146	121	60	6·8	17·45	2·68	20·13
	4800	$\frac{1}{2}$	All	148	128	42	8·5	16·54	1·72	18·26
			M.P. and L.P.	140	124	42	8·5	16·80	1·64	18·44
			None	139	127	42	8·5	16·45	1·75	18·20
	1600	$\frac{1}{6}$	All	122	76	25	11·5	17·60	3·40	21·00
			H.P. and M.P.	122	78·5	25	11·5	17·70	3·30	21·00
			None	126	80·5	25	11·5	19·20	3·30	22·50

Some important investigations upon the initial condensation in steam-engine cylinders have been carried out by Professor Capper for the Institution of Mechanical Engineers; but the small size of the engine used, cylinders $6\frac{1}{2}''$ and $11\frac{1}{4}''$ by 14 inches stroke, with a highest piston speed of 584 feet per minute, does not permit of any distinct conclusion as far as large marine engines are concerned. There is, however, abundant evidence from everyday naval practice to show that the conclusions drawn are in the main correct, and these may be summarised as follows:—

(1) Leakage past the slide valve appears to be independent of the speed of sliding surface, and to be proportional to difference of pressure between the two sides of the valve. The leakage is inversely proportional to the overlap of the valve, and with well-fitted valves the leakage might amount to 20 per cent of the steam entering the cylinder, and in no case less than 4 per cent. (This leakage is probably in the state of water, which may be partially re-evaporated on reaching the exhaust cavity of the valve. To prevent such leakage, separate valves for the steam and the exhaust have been suggested; but with stage expansion engines it is probable that the thermo-dynamic loss would be increased.)

(2) Without jacketing, with a given rate of expansion and allowing for slide-valve leakage, initial condensation *diminishes* while the total condensation *increases* with increased initial temperature.

(3) Re-evaporation for a given ratio of expansion is as great, and sometimes greater, without jackets as with them. This result tends to show that the regenerative action of the cylinder walls is largely independent of their mean temperature.

(4) There is a distinct temperature at which, for any particular speed of revolution and with a given ratio of expansion, jacketing becomes unnecessary and wasteful.

Superheated steam was extensively used to counteract cylinder condensation in marine engines when working pressures of from 5 to 35 lb. were common. With the advent of higher working pressures and, therefore, largely increased temperatures of steam, the difficulty of internal lubrication also greatly increased. (Naval reciprocating engines are now worked practically without internal lubrication; but the use of superheated steam makes internal lubrication imperative, and possibly decreases the boiler

efficiency.) With the higher flash-point oils, now used for internal lubrication, there is no difficulty in making lubrication efficient.

Dry steam does not quickly absorb heat which is supplied by the fuel, and consequently a much higher temperature must surround the tubes in which the steam is contained for the process of superheating than would otherwise be necessary, say, for water which absorbs heat very quickly. If the steam be superheated by the products of combustion after leaving the boiler and during their passage to the funnel, the ultimate temperature of the funnel gases must be high and a decreased efficiency results from the increased funnel loss. The early superheaters were fitted in this position, and when the temperature of the steam leaving the boiler seldom exceeded 280° F., a fairly large quantity of superheat could be added without increasing the funnel temperature (which under natural funnel draught only was generally above 600° F.), and without increasing the funnel loss. With the high pressures (300 lb.) and temperatures (422° F.) of steam now in common use, and the moderate funnel temperatures of 560° to 600° F. (frequently less with hot-air draught and with economisers), the difference which might be absorbed by the steam and used to superheat it is extremely small. It may be concluded, therefore, that although a superheater fitted between the boiler and the funnel might be, and in many instances was, productive of economy in early days, such a position would result in no practical economy to-day.

It follows that a superheater to be efficient must be placed where the temperature of the heated products of combustion is sufficiently high, and where the heat remaining, after effecting the process of superheating, can be efficiently used in heating the boiler water or in generating steam without increasing the funnel loss. In recent installations, therefore, the position of the superheater is either next the fire, where it is subjected to the greatest temperature, or in some intermediate position such as the combustion chamber of a cylindrical boiler or in a corresponding position in a water-tube boiler. In the mercantile marine, where a steady output of steam is maintained for long periods, superheating can be safely used without danger from a rise in temperature of the metal, generally mild steel, of the superheating tubes; in other words, the degree of superheat can be fairly well

regulated when the output of steam is practically constant, and the economy of fuel and efficiency of the steam may be increased. The same conditions obtain in many land installations of power, and a corresponding economy of from 5 to 25 per cent results.

Superheated Steam in Warships.—In warships the conditions are not in favour of superheating (although trials in a battleship showed economical results, and the possible economical advantages are well known and fully admitted), because the output of steam and the power developed by the engines are frequently varied when ships are in company. The rate of combustion should correspondingly vary, but the proper regulation of the amount of superheat becomes exceedingly difficult and almost impossible of attainment, and consequently the danger of overheating the superheater tubes is very great. Apart from this particular danger, the wear and tear of the internal mechanisms of the engine require, perhaps, still greater consideration, as these are continually subjected to a largely varying temperature. On the whole it may be stated that, in the conditions now obtaining, the reciprocating engines of warships are not likely to benefit permanently from any system of superheating and the use of superheated steam; and further, that even if economy might be gained under conditions favourable to its adoption, the extra weight and space and increased danger from overheating more than compensate any increased advantage. Increased economy by decreasing the funnel loss has, however, been obtained in other ways, principally exemplified by the Belleville and Hawthorn economisers, and by the still more recent adoption in naval vessels of the Howden system of forced draught with an accompanying economising device of hot-air supply.

With turbine engines internal lubrication is unnecessary, and therefore no difficulties arise from it. For naval vessels the difficulty of efficient regulation of the amount of superheat still applies, and necessitates an increased radial clearance between both the moving blades and the casing and between the fixed blades and the moving drum of the turbine. The larger clearances tend to increase the loss of steam (and energy), which escapes into the exhaust without doing useful work; but the rapid interchange of heat which takes place in the blade passages tends to counteract the clearance loss by absorbing moisture, and by maintaining a better quality of steam.

Stage Expansion.—Working expansively in a single cylinder is economical until a point is reached when the alternating temperatures in the cylinder become too great in range, so that a great part of the energy in the steam is lost through cylinder condensation and radiation from the internal and external surfaces of the cylinder. Generally a range of temperature of about 80° is the maximum that can be worked without serious loss; but at low pressures the losses are not so great, and ranges of from 100° to 120° are adopted with a fair economy. By dividing the expansion between two or more stages, in separate cylinders, the range of temperature in each is proportionately lowered, and this is done in practice by adopting stage-expansion, or compound, engines.

Economy, even with stage-expansion engines, can only be obtained when a proper range of temperature is adopted for each cylinder. Thus a two-stage engine, using 140 lb. pressure, shows little economy over one using 90 lb. pressure; while a three-stage engine shows a greater gain than theoretically due, when working at 140 lb. pressure against a two-stage engine working at 90 lb. The actual amount of fuel saved by using a three-stage engine working at 155 lb. pressure compared with a two-stage engine using 80 lb., is about 20 per cent.

For pressures over 180 lb. a four-stage engine is more economical in fuel than a lower-stage engine, and for mercantile purposes four stages are generally used for this economical reason. In the Navy, economy of weight and space are considered of more importance than a low fuel expenditure for full power, and a four-stage engine would be less economical at the low powers at which the bulk of the cruising is made.

Proportion of Cylinder Volumes.—The pistons have the same stroke in any one particular engine, and consequently the volumes of the cylinders are generally in exact proportion to their areas or the squares of the diameters of the cylinders; this last is the expression generally stated.

For naval engines the proportions adopted are as below:—

Date, about.	Initial Pressure.	H.P. Cyl.	M.P. Cyl.	L.P. Cyl.
1890	155 lb. per square inch	1	2·25	5 to 6
1894	250 ,, ,,	1	2·66	7·01
1903	210 ,, ,,	1	2·5	6·28
1905	250 ,, ,,	1	2·8	7·9

The cut-off is at about 0·75 of the stroke, and this produces an apparent ratio of expansion of about 7 to 10, but is actually less, as the clearance spaces are large.

For the mercantile marine the following proportions are common :—

Type of Engine.	Initial Pressure.	H.P. Cyl.	M.P. Cylinder.		L.P. Cylinder.
Two-stage expansion	90 lb.	1	...		5·2
,, ,,	120 ,,	1	...		6·75
Three-stage expansion	120 ,,	1	2·1		5·4
,, ,,	140 ,,	1	2·2		6·2
,, ,,	160 ,,	1	2·4		7·0
,, ,,	180 ,,	1	2·5		7·8
			1st M.P.	2nd M.P.	
Four-stage expansion	180 ,,	1	1·8	3·6	7·9
,, ,,	215 ,,	1	2·05	4·22	8·62 (*Orontes*)
,, ,,	215 ,,	1	1·73	4·0	9·0 (*Kaiser Wilhelm II.*)
,, ,,	267 ,,	1	2·0	4·0	12·2 (*Inchmarlo*)
,, ,,	210 ,,	1	2·0	3·9	7·9 (*Caronia*)

The cut-off adopted is generally at about 0·6 of the stroke, but there is some considerable variation in different engines.

Reason for adopting a Late Cut-Off in Naval Engines.—Naval engines are nearly always of the three-stage expansion type, and a late cut-off is adopted for full power. In the H.P. cylinder it is at about three-fourths stroke, and frequently the same cut-off is used for the M.P. and L.P. cylinders; it is seldom less than 0·625 of the stroke. This late cut-off is, of course, used at full power only, when the link is in full gear; at low powers, economy is obtained by "linking-up," thus obtaining an earlier cut-off, and a consequent higher ratio of expansion.

A late cut-off produces a higher mean pressure in the cylinder, and thus a greater power is obtained with a comparatively small engine. This is most important in warships, where the maximum power is only required for short periods on ordinary service, and the great bulk of the steaming is done at only 5 to 30 per cent of the full power. The machinery must be designed for the greatest stress to which it may be subjected. With a fairly constant pressure during the greater part of the stroke, obtained by a late cut-off, the turning moment becomes more uniform, and more nearly approaches the maximum: this results in a relatively large power for the

minimum weight. A more regular rotation of the propeller tends to steady its action and to produce freedom from vibration and fluctuations of speed. The power wasted in overcoming the frictional resistance is smaller in the relatively smaller engine with a late cut-off.

With engines on the four-stage expansion principle, the reasons given above still hold good, and no new principles are involved; but the power required for cruising is proportionately less compared with the increased full power now fitted to warships.

Comparative Size of Single and Three-stage Expansion Engines.—For purposes of design, the whole of the power is considered to be obtainable from the L.P. cylinder, and this would be the case in practice if cylinder condensation losses did not exist. The H.P. and M.P. cylinders are merely adjuncts of the L.P. cylinder for the purpose of dividing the expansion into stages, and thus saving the losses from condensation.

As an example of comparative size, consider two three-cylinder engines of the same power, length of stroke, rate of revolution, and nominal ratio of expansion; but one engine will exhaust from each cylinder directly into the condenser, while the other will take steam in series from cylinder to cylinder, and finally exhaust into the condenser. Then by a simple calculation it can be shown that if the diameter of each cylinder in the single-stage engine is 40 inches, the cylinders of the three-stage expansion engine will be about 28, 44, and 72 inches in diameter—that is, the higher-stage expansion engine occupies more space than the single-stage expansion engine of the same power; but as the coal consumption per I.H.P. is less in practice, the boiler power can be reduced, and generally a *total* saving of weight and space of machinery and boilers is obtained.

Efficiency of the Steam.—The table of maximum efficiencies of a perfect heat engine, given in Chapter V., shows that there is an increase—

1. By lowering the back or final pressure (and temperature).
2. By increasing the initial or boiler pressure (and temperature).
3. By superheating the steam before its admission to the engine (and thus increasing its temperature).

The increase of *actual* efficiency by the use of modern engines is much greater than would appear from the table, because the

efficiency practically obtained more nearly approaches the maximum efficiency.

The application of the indicator by which the indicated horse-power is obtained gives the only real measure of the efficiency of the steam. For example, consider an engine working with an initial pressure of 210 lb. per square inch (temperature 389° F.), and with a feed-water temperature of 100°; and that for each horse-power, as found by the application of the indicator, about 15 lb. weight of steam are required per *hour*. About 42·3 b.t. units are required for each horse-power, or 33,000 foot-lb. of work done per *minute*.

The useful work done by this 15 lb. weight of steam is (per hour)—

$$42{\cdot}3 \times 60 \text{ (minutes)} = 2538 \text{ b.t. units.}$$

The total heat supplied (net) per 60 minutes

$$= 15 \times \{966 - 0{\cdot}7(389 - 212) + (389 - 100)\}$$

$$= 15 \times 1131 = 16{,}975 \text{ b.t. units;}$$

and the *actual efficiency* of the steam $= \dfrac{2{,}538}{16{,}975} = 15$ per cent nearly.

A steam consumption of from 15 to 17 lb. per I.H.P. per hour is that usually obtained from warships; but in the mercantile marine it is frequently less—from 14 to 12 lb. only is fairly common. Calculated as above, the steam efficiency of a turbine using 11·49 lb. of steam per I.H.P.-hour

$$= \frac{42{\cdot}3 \times 60}{11{\cdot}49 \times 1131} = 20 \text{ per cent nearly.}$$

This may be considered as about the *maximum actual* efficiency of the steam at present attainable, and is about 59 per cent of the maximum efficiency of a steam engine considered as a perfect heat engine.

Summary of Heat Losses.—The following estimate of the actual losses of heat is taken from experimental results obtained from a good and fairly efficient steam plant. The figures given are based on the assumption of a supply of 100,000 b.t. units from the fuel burnt:—

Description of Loss.	Unpreventable Loss per cent.	Actual Engine per cent.	Approximate per cent.
Boiler radiation and leakage	...	5·450	5½
Radiation of flue gases on way to economiser	...	0·545	½
Radiation from economiser	...	2·725	2¾
Up the funnel	8·540	10·980	11
Radiation from feed pipe to boiler	...	0·135	¼
Total boiler losses	8·540	19·835	20
Steam pipes, radiation from	...	1·685	1½
Radiation from jacket drain pipes	...	0·083	...
Radiation from engine	...	2·452	2½
Engine friction	...	1·000	1
Total engine losses	...	3·535	3½
Radiation from eduction pipe	...	0·925	1
Radiation from feed arrangements	...	1·060	1
Heating the circulating water	65·220	59·150	59
Total condenser losses	65·220	61·135	61
Total losses	73·760	86·190	86
Leaving useful work	26·24	13·810	14

When temperature of steam at admission is 359° Fahr.
,, ,, ,, exhaust is 100° ,,
,, ,, feed water to economiser is 81° ,,
,, ,, ,, boiler is 278° ,,
,, ,, jacket water entering boiler is 328° ,,
,, ,, ,, leaving engine is 335° ,,

Various Efficiencies.—The figures in the above summary, although true in this particular experiment, are not conclusive evidence of the actual losses in any other instance. They can, however, serve the purpose of showing how the various efficiencies of an actual engine are calculated.

(1) The *total or overall* efficiency of the engine takes into account the engine friction, and therefore when 13,810 out of 100,000 b.t. units are converted into useful work—

$$\text{Efficiency} = \frac{13{,}810}{100{,}000} = 13{\cdot}81 \text{ per cent} \left(= \frac{\text{Shaft energy}}{\text{Fuel energy expended}} \right).$$

(2) 19,835 units are lost in various ways from the boiler, leaving 80,165 actually in the steam on its exit from the boiler. 1685 units are lost from the steam pipe before the steam reaches the engine, and thus leave a *net* supply of heat in the steam to the

engine of 78,480 units. Including the steam pipe losses with those in the boiler—

$$\text{Efficiency of the } \textit{Boiler} = \frac{78{,}480}{100{,}000} = 78{\cdot}48 \text{ per cent.}$$

(3) From the 78,480 units supplied to the engine in the steam, radiation from the jacket drain pipes and from the engine take away 2535 units and the condenser losses (or exhaust waste) absorb 61,135 units, thus making a total loss of heat in the steam of 63,670 units—

$$\text{Efficiency of the } \textit{Steam} = \frac{78{,}480 - 63{,}670}{78{,}480} = \frac{14{\cdot}81}{78{\cdot}48}$$
$$= 19 \text{ per cent nearly.}$$

(4) Of the 14,810 units converted into useful work on the pistons, 1000 are absorbed in engine friction—

$$\textit{Mechanical} \text{ efficiency of the } \textit{Engine} = \frac{14{\cdot}81 - 1}{14{\cdot}81} = 93{\cdot}24 \text{ per cent.}$$

(5) The total or overall efficiency of the engine is the product of the *three* efficiencies of the boiler, the steam, and the mechanism respectively—

$$= \frac{78{\cdot}48}{100} \times \frac{14{\cdot}81}{78{\cdot}48} \times \frac{13{\cdot}81}{14{\cdot}81} = 13{\cdot}81 \text{ per cent, which agrees with (1).}$$

(6) The *maximum efficiency, considered as a perfect heat engine,* does not take into account the boiler and engine friction losses, but only those of the steam as a working substance between its temperature limits. This efficiency is thus only comparable with the efficiency of the steam, as given in (3)—

$$\text{Maximum efficiency} = \frac{T - t}{T + 461} = \frac{359 - 100}{359 + 461} = \frac{259}{820}$$
$$= 31{\cdot}6 \text{ per cent nearly.}$$

(7) Compared with the maximum efficiency, the steam efficiency of the above engine—

$$= \frac{19}{31{\cdot}6} = 60 \text{ per cent about of the maximum possible efficiency.}$$

(8) The *thermal efficiency* of a steam plant—

$$= \frac{\text{heat converted into indicated work}}{\text{heat supplied to boiler}},$$

which in the above example $= \frac{14{,}810}{100{,}000} = 14{\cdot}81$ per cent.

The thermal efficiency is the product of the boiler and steam efficiencies, and in practice is generally calculated from the coal or fuel used per I.H.P. per hour, when the calorific value of the fuel is known.

EXAMPLE.—An engine uses $1\frac{1}{2}$ lb. of coal per I.H.P.-hour, and the calorific value of the coal is 15,000 b.t. units per lb.—

$$1 \text{ I.H.P.-hour} = \frac{33{,}000}{780} \times 60 \text{ (minutes)} = 2538 \text{ b.t. units per hour.}$$

$$\text{Thermal efficiency} = \frac{2538}{1\frac{1}{2} \times 15{,}000} = 11{\cdot}28 \text{ per cent.}$$

Comparative Thermal Efficiencies of Heat Engines.—Taking the formula for the maximum efficiency of a perfect heat engine as a basis of comparison :—

A *steam* engine receiving steam at a temperature of 486° F., and exhausting it at 113° F. into a condenser—

$$\text{Efficiency} = \frac{486 - 113}{486 + 461} = \frac{373}{957} = 38{\cdot}8 \text{ per cent.}$$

A *gas* or *oil* engine working between the limits of 2800° F. and 60° F. (exhausting into the atmosphere)—

$$\text{Efficiency} = \frac{2800 - 60}{2800 + 461} = \frac{2740}{3261} = 84 \text{ per cent.}$$

Of which in each case about one-half may be realised in the best practice.

Efficiency of the Mechanism.—Recently the invention of the flash-light and other torsion meters has enabled engineers to study mechanism efficiency from an entirely new and reliable stand-point. Previously the method of estimating shaft H.P. was almost entirely empirical, and based on generalities and theoretical estimation rather than on any really sound basis. One of the most instructive

matters in connection with mechanical efficiency is described below, and reference should be made to shaft H.P. in Chapter V., and to certain theoretical considerations on the pressure on the guide at the end of Chapter XV.

It is now proposed to compare the crank effort produced by the actual effective pressure in the cylinder with that observed by a torsion meter. In Fig. 272 the crank circle is divided into 24 equal parts, like a 24-hour (continental) clock, and the actual crank effort, or turning moment, at each point is then plotted on a curve, shown in full lines in Fig. 272A, for the L.P. cylinder. Suppose the crank is in any position, AC, and the connecting rod is then represented by BC, produced. Then, the turning moment, or crank effort, at C = force exerted along $BC \times AM$. The effective pressure in the cylinder can be obtained from the indicator diagram, as shown, in which the various positions of measurement (corrected for obliquity) are made to correspond with the position of the piston in the cylinder.

Let the effective pressure at position **2** be represented by P in lb., and of this force a certain proportion is exerted on the guide so that, graphically represented, the proportion acting in the line $BC = \frac{BM}{AB} \times P. = \frac{BC}{BD} \times P.$ (where CD is perpendicular to BC).

The crank effort at **2**, therefore,

$$= P \times \frac{BC}{BD} \times AM.$$

By adopting a connecting-rod length of 4 cranks, the usual practice, a table of values at each of the 24 points for BD and AM can easily be formed, and the curve shown in Fig. 272A can be plotted. The one shown in the figure has been obtained by graphic measurement and slide-rule calculation, and may not be exactly accurate, but it shows clearly the salient points. Two dotted curves are also shown which represent the crank efforts of the H.P. and M.P. respectively; the three cranks being disposed at equal angles of 120 degrees, the three curves are plotted correspondingly. The combined efforts on the three cranks are next added together and a new curve set up from the same base line; this curve is the upper one shown in full lines. The average effort so found is shown by the straight full line parallel to the base. From these curves it

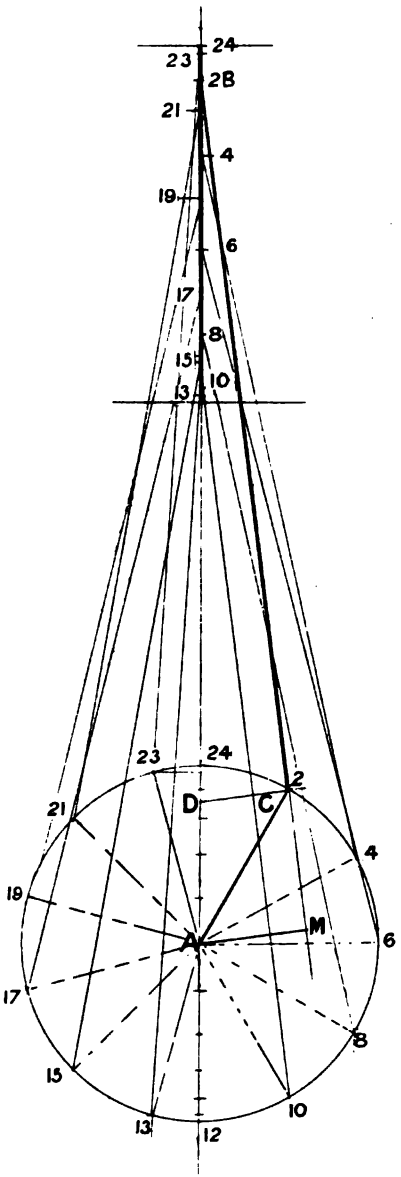


FIG. 272.—Calculation of Crank Effort from Indicator Diagram.

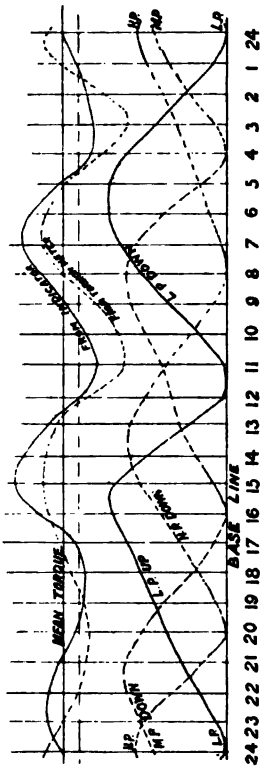
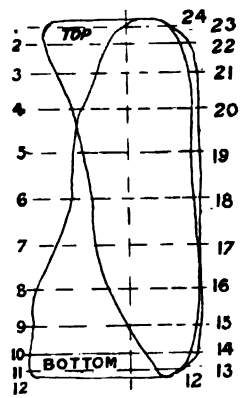


FIG. 272A.—Curves of Crank Effort, or Torque.



Indicator Diagram.

will be seen that the combined effort of a three-crank engine is by no means uniform, and this is true of any expansive working engine with any number of cranks.

Below the upper curve another is shown in dotted lines, corresponding to the actual torque, as determined by the flash-light torsion meter, for the same engine working under the same conditions of power and revolution. From this curve it will be seen that still greater variation (maximum and minimum) is obtained than appears in the indicator measurement curve. The average torque is shown by the straight dotted line. The difference in the heights of these two straight lines above the base line represents the proportion (about $8\frac{1}{4}$ per cent) of the I.H.P. expended in overcoming the frictional resistances of the engine mechanism, which in this particular case includes the thrust friction, because the shaft H.P. torque was measured abaft the thrust block.

For a turbine engine the torque is practically uniform, and any variation points to defective balancing of the rotating parts or to unequal work done by the several blades of the propeller. Incidentally, the various curves show that equal powers transmitted through identical shafting produce considerably less maximum stress when the shaft is rotated by a turbine than when rotated by a reciprocating engine, however well it may be proportioned to obtain as uniform a turning moment as possible. With a reciprocating engine the fluctuation of torque produces corresponding variations in the twist of the shaft during every revolution and thus tends to reduce the life of the shaft by continual change in stress and shape, which may eventually end by fracture.

In the table given below the port L.P. turbine shows less shaft horse-power than the starboard L.P. On the opening up, it was found that the tip clearances of the blades were slightly different, and sufficient to account for the differences in power.

[TABLE

FLASH-LIGHT TORSION METER

ACTUAL READINGS AND CORRESPONDING HORSE POWERS TAKEN DURING TRIAL TRIPS UNDER VARYING CONDITIONS OF DISPLACEMENT AND PROPELLERS

Turbine Shaft.	Degrees Torque.	Revs. per Min.	Shaft Horse-Power.	S.H.P. Total.
Starboard L.P. .	1·43	482·9	2775	
Centre H.P. . .	1·69	461·2	2600	7975
Port L.P. . .	1·37	472·8	2600	
Starboard L.P. .	1·32	461·2	2410	
Centre H.P. . .	1·65	426·8	2330	6940
Port L.P. . .	1·24	457·3	2200	
Starboard L.P. .	1·15	426·4	1970	
Centre H.P. . .	1·52	417·6	2080	5960
Port L.P. . .	1·13	418·9	1910	
Starboard L.P. .	1·05	418·4	1765	
Centre H.P. . .	1·52	422·3	2120	5555
Port L.P. . .	1·02	415·5	1670	
Starboard L.P. .	·21	198·6	162	
Centre H.P. . .	·27	206·3	185	495
Port L.P. . .	·19	183·5	148	
Starboard L.P. .	·22	146·7	88	
Centre H.P. . .	·21	171·4	87	257
Port L.P. . .	·13	144·8	82	
Starboard L.P. .	·07	46·3	13	
Centre H.P. . .	·05	86·1	15	37·2
Port L.P. . .	·01	24·4	9·2	

Coal Consumption.—The average results of many engines are given below, but as each individual ship varies its consumption from day to day, only a general conclusion can be drawn. This is, however, very definite in relation to the decreased consumption for increasing pressures and stages of expansion :—

I.H.P. PER 1 TON OF COAL CONSUMED PER HOUR, ALL PURPOSES .

Type of Engine.	Mercantile Marine.	Royal Navy.
Single-stage expansion	740	720 to 740
Two-stage expansion	1120	850 to 1000
Three-stage expansion	1240 to 1400	1000 to 1240
Four-stage expansion	1450 to 2000	.....

The following are some recent examples of battleships, taking an average of six on their trials:—

	One-fifth Power.	Four-fifths Power.	Full Power.
I.H.P. per ton of coal consumed per hour, *all purposes* . . .	1031	1120	1108
Speed of ship, in knots	11·9	17·8	18·92
Average for four first-class cruisers of 30,000 I.H.P.			
I.H.P. per ton of coal consumed per hour, *all purposes* . . .	1256	1240	1193
Speed of ship, in knots . . .	15·48	22·02	23·2

Since these cruiser trials were run, propellers with increased blade area have been fitted, and another knot of speed added at full power.

TABLE

Fuel burnt per I.H.P. per Hour.	I.H.Power developed per 1 Ton of Fuel burnt per Hour.									
lb.	0·00	0·01	0·02	0·03	0·04	0·05	0·06	0·07	0·08	0·09
0·8	2800	2766	2732	2696	2664	2632	2602	2574	2544	2512
0·9	2480	2456	2432	2410	2386	2358	2334	2308	2284	2262
1·0	2240	2217	2194	2172	2150	2133	2113	2093	2094	2056
1·1	2037	2019	2000	1983	1966	1048	1932	1916	1900	1884
1·2	1867	1852	1837	1822	1807	1792	1778	1764	1750	1736
1·3	1723	1720	1707	1695	1683	1660	1648	1636	1624	1612
1·4	1600	1589	1578	1567	1556	1545	1534	1523	1512	1502
1·5	1493	1483	1473	1464	1454	1445	1435	1426	1417	1408
1·6	1400	1392	1383	1374	1366	1357	1348	1340	1332	1324
1·7	1316	1308	1301	1294	1287	1280	1272	1264	1256	1248
1·8	1240	1234	1228	1222	1216	1211	1205	1199	1193	1186
1·9	1179	1173	1167	1161	1154	1148	1142	1136	1131	1125
2·0	1120	1114	1108	1105	1099	1093	1087	1081	1076	1071
2·1	1066	1061	1058	1050	1046	1041	1036	1031	1026	1020
2·2	1016	1014	1009	1004	1000	995	990	986	982	978
2·3	974	969	965	961	957	953	949	945	941	937
2·4	933	929	925	921	917	914	910	906	902	899
2·5	896	892	888	885	882	878	875	871	868	865
2·6	861	857	854	851	848	845	841	838	835	832
2·7	829	826	923	820	817	814	811	808	805	802
2·8	800	797	794	791	788	786	783	780	777	774
2·9	772	769	766	764	762	759	756	753	751	748
3·0	746	743	741	738	736	734	732	730	728	726

Coal Consumption and Speed.—The following table, compiled from one of the four cruisers above, gives some idea of the cost of high speeds. The results were obtained in actual service. It will be noticed that the last two knots cost about 8500 I.H.P.; an extra $9\frac{1}{3}$ tons of coal are burnt per hour, while the extra weight of machinery and boilers may be estimated at 650 to 700 tons.

Speed.	I.H.P.	Coal burnt per 24 Hours for all purposes.	Distance run per Ton of Coal burnt.	Percentage of Full Power (80,000 I.H.P.).
Knots.		Tons.	Knots.	
10	2100	73	3·2	7
12	3650	104	2·75	12·2
14	5700	158	2·12	19·0
16	8100	216	1·77	27·0
18	11,800	294	1·46	39·7
20	16,000	396	1·21	53·3
22	22,500	526	1·00	75
24	31,000	750	0·8	103·6

CHAPTER XXIII

ELEMENTARY THERMO-DYNAMICS

In the early chapters some idea is given of the nature and properties of heat, and its general relation to the working of the steam engine. In the present chapter it is proposed to carry this idea a step further and to develop certain details relating to the internal economy of heat engines, including the compression of air, compressed-air engines, and steam engines in general. The symbols which are used have been purposely chosen as being those usually found in works on Heat and Heat Engines, and, for convenience of reference, are tabulated below :—

r	Ratio of expansion.
e	Napierian base of logarithms.
T_0, t_0	Absolute temperatures.
P, p	Absolute pressures in lb. per 1 square *foot*.
Pm	Mean absolute pressure „ „
V, v	Volumes, in cubic feet, of 1 lb. of the substance.
Kp	Specific heat at constant pressure.
Kv	Specific heat at constant volume.
y	Ratio of the above specific heats $= \frac{Kp}{Kv}$.
c	Constant for a particular substance under discussion.
S	Sensible heat per 1 lb. of steam.
L	Latent heat of evaporation per 1 lb. of steam.
R	Internal latent heat of evaporation of steam.
x	Dryness fraction of steam.
H, h	Total heat of formation of 1 lb. of steam.

Summary of Previous Formulae, etc., relating to heat and heat engines :—

Boyle's Law : $V \propto \frac{1}{P}$, when T_0 is constant,

Charles' Law: $P \cdot V = c \cdot T_0$.

External work done $= p \cdot (V - v)$ (in ft.-lb. for each 1 lb. of the working substance).

Sensible heat: $S = T - t$. per 1 lb. of the working substance.

Maximum efficiency of an engine, using a perfect gas as the working substance only—

$$= \frac{T_0 - t_0}{T_0}.$$

Indicated horse-power developed in a single cylinder $= \frac{2 \cdot PLAN}{33{,}000}$

(with 2N working strokes per minute).

First law of thermo-dynamics:

Mechanical equivalent of heat, $J = 780$ ft.-lb. $= 1$ British thermal unit.

Second law of thermo-dynamics:

$$\text{Efficiency of conversion} = \frac{\text{heat converted into work}}{\text{heat taken in by the engine}}.$$

Latent heat of evaporation: $L = 966 - 0{\cdot}7(T - 212)$, per 1 lb. of saturated steam.

Total heat of evaporation: $H = 966 - 0{\cdot}7(T - 212) + (T - t)$
$= L + S$, per 1 lb. steam.

Total heat of formation of 1 lb. of steam, when dryness fraction $= x$:

$$= x \cdot L + S$$
$$= x \cdot \{966 - 0{\cdot}7(T - 212)\} + (T - t).$$

Equivalent evaporation, *from* and *at* 212° F. $= \frac{n \cdot (x \cdot L + S)}{966}$, where $n =$ lb. weight of steam at a given temperature, T, produced from a known temperature (t) of feed water.

Expansion of steam:

Hyperbolic curve, $P \cdot V =$ constant (T changes in value).

Saturation curve, $P \cdot V^{1{\cdot}0645} =$ constant.

Adiabatic curve, $P \cdot V^{1{\cdot}135} =$ constant, for steam initially dry and saturated.

$P \cdot V^{1{\cdot}035 + \frac{x}{10}} =$ constant, for wet steam of dryness fraction x.

Work done during Hyperbolic Expansion.—In Fig. 270 the area of *ADEQB* can be represented by some rectangular area,

ARSB, when the height *AR* or *BS* is equivalent to the mean height of *DEQ* or the mean pressure; call this p_m.

Let AD be called p_1 = initial pressure (absolute).

AB be called r = ratio of expansion

$$= \frac{\text{AB}}{\text{AC}}, \text{ when C is the point of cut-off.}$$

Then the work done during admission = area ADEC

$$= p_1 \times \text{AC},$$

and the work done during expansion = area CEQB

$$= \int_1^r p \times dv.$$

Now, $p \,.\, v = \text{constant} = p_1 \times 1$, for unit volume.

And, therefore, total work done $= p_1 \times 1 + \int_1^r \frac{dv}{v} \,.\, p_1$

$$= p_1 + p_1 \log_e r$$

$$= p_1(1 + \log_e r).$$

$$\text{Now, } p_m = \frac{\text{area ARSB}}{\text{AB}} = \frac{\text{area ADEQB}}{r}$$

$$= \frac{\text{work done}}{r}.$$

$$\text{Therefore } p_m = p_1\left(\frac{1 + \log_e r}{r}\right).$$

Fig. 270 also illustrates how economy can be obtained—

(*a*) With increased expansion. If the rate of expansion is increased, represented by *AY* instead of *AB*, then the area representing the work done during expansion is *CEZY* instead of *CEQB*, an increase without the expenditure of more steam.

(*b*) With increased pressure. If the initial pressure is increased, represented by *AF* instead of *AD*, and the area *AFGk* is made equal to *ADEC* (the work done during admission), there is the same work done during admission; but there is an increased amount, *kGQB*, done during expansion instead of *CEQB*, an increase with a very slightly greater expenditure of heat to obtain the higher initial pressure.

The increase in initial pressure does not increase the size of

cylinder necessary for the same rate of expansion, although more work can be done by the increased pressure.

The increase of the rate of expansion without increasing the initial pressure, although it increases the work done, involves the use of a larger cylinder to carry out the expansion; in other words, an increase in size of engine is necessary to obtain greater economy unless accompanied by an increase of pressure.

Specific Heat.—The specific heats of the solid and liquid substances commonly used in marine engineering practice are given in Chapter III. In the following table the specific heats of some of the principal gases are now shown :—

Name of Gas.	Specific Heat at Constant Pressure, K_p.	Specific Heat at Constant Volume, K_v.	Ratio of Specific Heats, $\gamma = \frac{K_p}{K_v}$.
Air, dry	0·238	0·169	1·4
Carbon dioxide, CO_2	0·217	0·172	1·26
Carbon monoxide, CO	0·245	0·173	1·41
Hydrogen, H_2	3·409	2·412	1·41
Marsh gas, CH_4	0·593	0·47	1·26
Nitrogen, N_2	0·244	0·173	1·41
Olefiant gas, C_2H_4	0·404	0·359	1·12
Oxygen, O_2	0·218	0·155	1·4
Products of combustion—			
(Funnel gases)	0·24	0·17	1·4 } Considerable variation in these values.
(Coal gas engine)	0·258	0·189	1·37 }
Steam, superheated and above 212° F.	0·48	0·37	... }

A gas has two specific heats: the specific heat at constant *pressure* and the specific heat at constant *volume*. The general idea may best be illustrated by examples in common practice :—

1. *When under Constant Pressure.*—In boiler furnace combustion the pressure of the air and of the products of combustion is very nearly atmospheric, and may be taken as constant: the specific heat at constant pressure of the products may be taken as about 0·25.

Suppose that 1 lb. of fuel gives out 15,000 b.t. units when burnt, and requires 19 lb. of air for its combustion in a furnace; then the products of combustion weigh $1 + 19 = 20$ lb., and the heat given out by combustion is distributed among these 20 lb.

The rise of temperature of each 1 lb. of products for each 1 b.t.u.

$$= \frac{1}{0{\cdot}25} = 4^\circ \text{ F.}$$

The rise of temperature of 20 lb. of products for 15,000 b.t.u.

$$= \frac{15{,}000}{20 \times 0{\cdot}25} = 3000^\circ \text{ F. about.}$$

2. *When under Constant Volume.*—In a closed cylinder the volume cannot increase and is, therefore, constant; the specific heat at constant volume may be taken as about 0·19, when liquid fuel is used as the combustible.

Suppose that 1 lb. of liquid fuel gives out 19,000 b.t. units and requires 14 lb. of air for its combustion; then the products of combustion weigh $1 + 14 = 15$ lb., and the heat given out by combustion is distributed between these 15 lb. of products.

The rise in temperature of each 1 lb. of products for each 1 b.t.u.

$$= \frac{1}{0{\cdot}19} = 5{\cdot}26^\circ \text{ F.}$$

The rise in temperature of 15 lb. of products for 19,000 b.t.u.

$$= \frac{19{,}000}{15 \times 0{\cdot}19} = 6667^\circ \text{ F.}$$

Definitions.—Specific heat at constant pressure (Kp) is the quantity of heat taken in by 1 lb. of a substance when its temperature rises 1° F., while the pressure remains unchanged and the volume is allowed to change (Fig. 273); or

$$\text{K}p = p \,.\, (\text{V} - v).$$

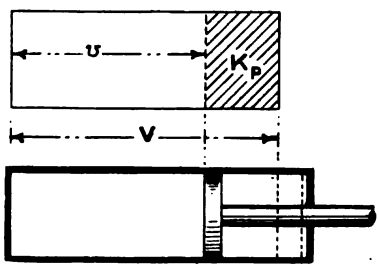


FIG. 273.—Value of Kp.

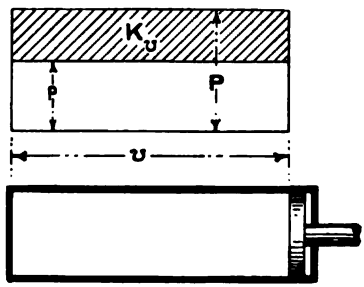


FIG. 274.—Value of Kv.

Specific heat at constant volume (Kv) is the quantity of heat taken in by 1 lb. of a substance when its temperature rises 1° F.,

while the volume remains unchanged and the pressure is allowed to change (Fig. 274); or

$$Kv = v(P - p).$$

Laws of the Permanent Gases. Third Law (Regnault).—The specific heat at constant pressure (Kp) is constant for any gas.

The approximate truth of this law has been demonstrated by experiment, and simply means that the heat taken in by 1 lb. of the substance is the same whether the rise of temperature be from, say, 32 to 33°, or from, say, 400 to 401° F., so long as the difference is exactly 1° F.

At constant pressure, the external work done due to a change of volume from v to $V = p.(V - v)$; and if T and t be the corresponding temperatures, the heat taken in by 1 lb. of the substance $= Kp.(T - t)$.

The increase in internal energy $= Kp.(T - t) - p.(V - v)$.

From Charles' Law, $p.V = c.T_0$, and $p.v = c.t_0$;

and thus $p(V - v) = c(T_0 - t_0)$.

Therefore, increase in internal energy per 1 lb. of the substance

$$= Kp.(T - t) - c.(T - t)$$
$$= (Kp - c)(T - t).$$

That is, the change in internal energy is exactly proportional to the change of temperature, when the pressure is constant.

At constant volume (when no external work is done) any heat taken in by the substance produces a change in temperature and pressure, and $P \propto T_0$. The heat taken in by 1 lb. of the substance $= Kv.(T - t)$, and this must evidently = the increase in internal energy of each 1 lb. of the substance. That is, the change in internal energy is exactly proportional to the change of temperature, when the volume is constant.

Therefore, when the temperature of a substance is raised from t to T, the increase in internal energy at constant pressure = the increase in internal energy at constant volume; and

$$(Kp. - c)(T - t) = Kv.(T - t)$$

From which, $$Kv = Kp - c,$$

and $$Kp - Kv = c.$$

Fourth Law (Joule).—When a gas expands without doing external work, and without either taking in or giving out heat (and,

therefore, without changing its intrinsic energy), its temperature does not change.

This is a continuation of the application of Boyle's Law—

$$p \,.\, v = \mathrm{P} \,.\, \mathrm{V} = \text{constant, when T is constant.}$$

From the third law, it may now be written:

$$\frac{p \,.\, v}{t_0} = c = \frac{\mathrm{P} \,.\, \mathrm{V}}{t_0} = c = \mathrm{K}p - \mathrm{K}v.$$

The fourth law, like the others, is not exactly true except of a perfect gas. The difference is extremely small, but is distinctly apparent in the adiabatic expansion of steam particularly, and of air; in each case there is a less pressure at the end of the expansion than would be shown by exact calculation based on the absolute truth of the law as enunciated.

Values of Kp, Kv, and y. For *dry air*:—
As measured by Regnault, the value of

$$\mathrm{K}p = 0{\cdot}2375 \text{ b.t. units.}$$

Assuming that 1 b.t.u. $= 780$ ft.-lb.; and $c = 53{\cdot}18$ (p. 26).
Then $\mathrm{K}p = 184{\cdot}7$ ft.-lb.
From which, $\mathrm{K}v = \mathrm{K}p - c = 184{\cdot}7 - 53{\cdot}18$
$= 131{\cdot}5$ ft.-lb $= 0{\cdot}1696$ b.t. units.

And,
$$y = \frac{\mathrm{K}p}{\mathrm{K}v} = \frac{\text{specific heat at constant pressure}}{\text{specific heat at constant volume}}$$
$$= \frac{0{\cdot}2375}{0{\cdot}1696}$$
$$= 1{\cdot}4 \text{ about.}$$

For *superheated steam*, whose density under similar conditions of pressure and temperature is only 0·622 times that of dry air, the different value for c becomes—

$$= \frac{c}{0{\cdot}622} = \frac{53{\cdot}18}{0{\cdot}622} = 85{\cdot}5 \text{ ft.-lb.;}$$

and, therefore, $\mathrm{P} \,.\, \mathrm{V} = 85{\cdot}5 \ \mathrm{T}_0$ for superheated steam (see p. 44), where T must be some temperature above that corresponding to the boiling-point of saturated steam at the pressure considered.

Other values for superheated steam are—

$$Kp = 0{\cdot}48 \text{ b.t. units} = 374{\cdot}4 \text{ ft.-lb.}$$
$$Kv = 374{\cdot}4 - 85{\cdot}5 = 288{\cdot}9 \text{ ft.-lb.}$$
$$y = \frac{Kp}{Kv} = 1{\cdot}296 \qquad = 1{\cdot}3 \text{ nearly.}$$

General Law of Expansion of Gases, when doing Work.— When any small increment of heat, dH, is supplied to a substance, there is a gain in intrinsic energy which may or may not be converted into work. In any case—

$$d\mathrm{H} = Kv \,.\, dt_0 + p \,.\, dv.$$

When the expansion is adiabatic, no heat being received into the substance, the work done = change in intrinsic energy of the substance, and—

$$d\mathrm{H} = 0, \text{ or zero.}$$

Therefore $$Kv \,.\, dt_0 + p \,.\, dv = 0 \quad . \quad . \quad . \quad (1).$$

For a perfect gas, $$p \,.\, v = c \,.\, t_0.$$

From which, $p \,.\, dv + v \,.\, dp = c \,.\, dt_0$. But $c = Kp - Kv$;

therefore $$dt_0 = \frac{p \,.\, dv + v \,.\, dp}{Kp \,.\, - Kv}.$$

Substituting this value in (1) for dt_0—

$$\frac{Kv}{Kp - Kv}(p \,.\, dv + v \,.\, dp) + p \,.\, dv = 0.$$

That is, $$Kv \,.\, v \,.\, dp + Kp \,.\, p \,.\, dv = 0.$$

Divide by $Kv \,.\, p \,.\, v$, then $$\frac{dp}{p} + \frac{Kp \,.\, dv}{Kv \,.\, v} = 0.$$

Substituting y, and integrating—

$$y \,.\, \log v + \log p = \text{constant};$$

from which, $$p \,.\, v^y = \text{constant.}$$

With a certain mass of dry air contained behind a piston in a cylinder, the temperature remains constant if the piston be moved by some external agency and allows the air to expand without doing work. The energy contained in the mass of air undergoes no change—it is always equal to P.V.; thus following Boyle's Law and also that of Charles, because T does not change.

Again, if work be done while the piston moves, or, say, that the pressure of air overcomes a resistance opposed to the piston, then energy in the form of heat is taken from the mass of the gas. The removal of this heat, and its conversion into mechanical energy or energy of motion, leaves the mass of the air at a lower temperature. The drop in temperature, if the pressure remain constant, would itself produce a less volume; consequently, as the volume is allowed to increase also, the pressure at the end of the expansion is less than in the case where no work is done. The form of the expansion curve now follows that given by the expression, $P.V^n =$ constant, in which n is always greater than unity (1) and generally less than 1·4.

Air Compression and Compressed-air Engines.—A simple graphical illustration of a cylinder and piston working under the stated conditions shows fairly clearly the result which may be

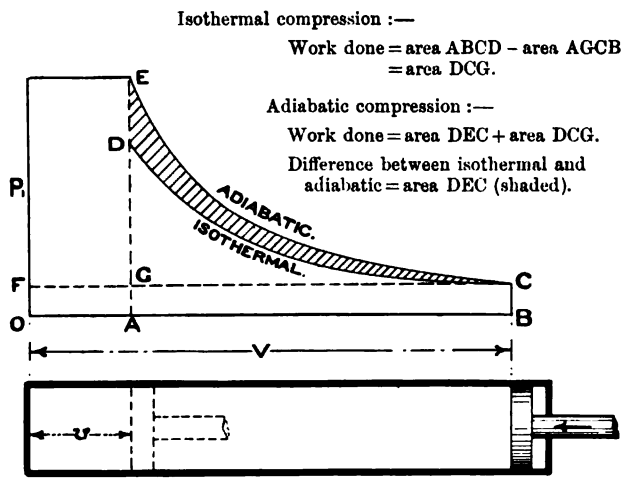


FIG. 275.—Work done during Compression of Dry Air.

expected when mechanical energy is converted into heat energy by compressing air (Fig. 275), or when heat energy is converted into mechanical energy by compressed air driving a piston before it and when the previous conversion of energy is reversed (Fig. 276).

In a submerged torpedo driven by compressed-air engines, *without supplementary heating*, the temperature of the working cylinders is kept nearly constant by the heat supplied from the surrounding water. Without this supply the engines might possibly

be chilled and any moisture present in the air become frozen. Consequently, because the temperature of the air is higher than it would otherwise be at the end of the expansion, the pressure on the piston is better maintained during the expansion and more work is given out by the engine. In actual practice the importance of

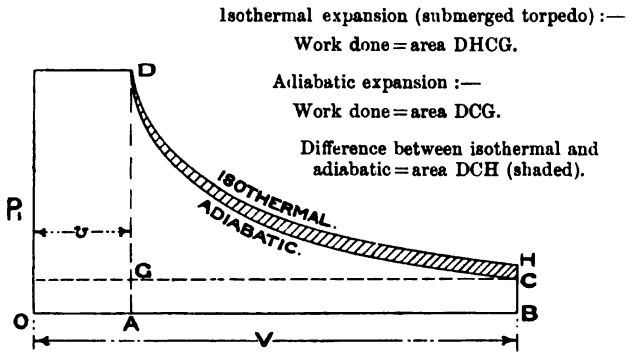


FIG. 276.—Work done during Expansion of Dry Air.

obtaining a temperature of the air in the torpedo storage-chamber as nearly as possible equal to that of its surroundings, is thus emphasised. If the temperature of the air be above that of the surrounding water, the pressure is decreased and the power decreased also. If the temperature be below that of the surrounding water (an almost impossible condition), the air in the chamber tends to expand and, being restricted in its volume, the pressure is increased (to perhaps a dangerous extent in an extreme case).

The highest practical efficiency of the machine in which air is compressed can only be obtained by a minimum rise in temperature of the air so compressed. This is fairly evident if we consider that any rise in temperature produces an equivalent increase in pressure which the machine has to overcome; and in doing so, more work must be done than if the temperature had remained approximately constant. Represented graphically, the extra work required is shown by the area contained between the two curves of the isothermal and the actual compression (Fig. 275).

In a purely suppositious case, when no cooling is effected either by water cooling or radiation, etc., a large amount of the work done by the machine is transformed into heat and the temperature of the

air is considerably raised. The rise in temperature is obtained as follows :—

Let T_0 be the final temperature, V be the corresponding volume, and P be the pressure.

Let t_0 be the initial temperature, v be the corresponding volume, and p be the pressure.

And let $y = \frac{Kp}{Kv} = 1{\cdot}4$ for air.

Then from previous sections we have—

$$P \cdot V^y = p \cdot v^y \quad \text{and} \quad \frac{p \cdot v^y}{P \cdot V^y} = 1.$$

Also $\frac{T_0}{t_0} = \frac{P \cdot V}{p \cdot v}$, approximately, air being nearly a perfect gas.

From which,

$$\frac{T_0}{t_0} = \frac{P \cdot V}{p \cdot v} \times \frac{p \cdot v^y}{P \cdot V^y} = \frac{v^{y-1}}{V^{y-1}} = \left(\frac{v}{V}\right)^{y-1}.$$

It is more convenient to measure the relative pressures than the relative volumes, and therefore we proceed to get pressures in place of volumes.

$$P \propto \frac{1}{V^y} \text{ and } p \propto \frac{1}{v^y}.$$

Therefore $\frac{P}{p} = \left(\frac{v}{V}\right)^y$ and $\frac{v}{V} = \left(\frac{P}{p}\right)^{\frac{1}{y}}$.

From which, $\frac{T_0}{t_0} = \left(\frac{v}{V}\right)^{y-1} = \left(\frac{P}{p}\right)^{\frac{y-1}{y}} = \left(\frac{P}{p}\right)^{\frac{0{\cdot}4}{1{\cdot}4}} = \left(\frac{P}{p}\right)^{\frac{2}{7}}$.

As already pointed out above, such actual temperatures are seldom, if ever, reached in ordinary practice; but an example of what might occur is instructive.

EXAMPLE.—A chamber is charged to 2250 lb. pressure per square inch, and the initial temperature of the air is 99° F. (= 560 absolute F.); find the rise of temperature *if no cooling is allowed.*

$$\frac{T_0}{t_0} = \left(\frac{2250}{14{\cdot}7}\right)^{\frac{2}{7}} = (153)^{\frac{2}{7}}.$$

Taking logarithms—

$$\text{Log } T_0 - \log t_0 = \tfrac{2}{7} . \log 153 \qquad \text{Log } 153 = 2{\cdot}1846914$$
$$= 0{\cdot}6241975. \qquad 2$$
$$T_0 = t_0 \times 4{\cdot}208 \qquad 7)\ 4{\cdot}3693828$$
$$= 560 \times 4{\cdot}208 \qquad \text{Log } 4{\cdot}208 = 0{\cdot}6241975$$
$$= 2356{\cdot}48.$$
$$T_0 - t_0 = 2356 - 560 = 1696^\circ \text{ F. rise of temperature.}$$

When air is compressed in an actual compressor only a moderate rise in temperature takes place, because there is always some radiation, etc., even when the pressure is pumped up very quickly; in addition, water circulation is used to cool the air both during compression and between the stages of compression. In the matter of more efficient cooling great improvements have been made in recent years, and economy in power has been obtained in consequence.

Work done during the Isothermal Compression of Dry Air.—The compression being isothermal, the compression curve is hyperbolic and $V \propto \frac{1}{P}$, because T is constant. The problem comes under external work, and the work done, represented by the area DCG of Fig. 275, $= p_m(V - v)$.

EXAMPLE.—If 105 lb. weight of air, of absolute pressure 15 lb. and temperature 102° F., be compressed isothermally in a chamber to 2400 lb. pressure per 1 square inch; find the work done, and the horse-power usefully expended in air compression, if the time required to so compress it is 80 minutes.

The work done $= p_m . (V - v)$, where $p_m = p_1 \times \frac{\log_e r}{r}$.

$p_1 = 2400 \times 144$ (lb. per 1 sq. foot).

$r = \frac{2400}{15} = 160.$

V = Volume of 105 lb. at 102° F. (= 563° abs. F.)

$= \frac{105 \times 12{\cdot}38 \times 563}{493}$ (1 lb. of dry air at 32° F. fills 12·38 cub. ft.)

$= 1485$ cubic feet.

v = Volume of same mass of air under 2400 lb. pressure per 1 sq. in.

$= \frac{1485 \times 15}{2400} = 10$ nearly.

$V - v = 1485 - 10 = 1475.$

$$\text{Work done} = \frac{2400 \times 144 \times 1475 \times 4{\cdot}6}{160} \text{ ft.-lb.} \qquad (\text{Log. } 160 = 4{\cdot}6052.)$$

$$\text{Horse-power (for 1 minute)} = \frac{\text{work done}}{33{,}000 \times 80}$$

$$= \frac{2400 \times 144 \times 1475 \times 46}{33{,}000 \times 80 \times 1600}$$

$$= 5\tfrac{1}{2} \text{ about.}$$

This result is, of course, below the actual I.H.P. of the engine, because some rise of temperature takes place in each stage of compression of the pump, and work is done also in overcoming the friction of the working parts of the engine. In actual practice, the I.H.P. of the engine may be taken as about *four* times the horse-power as calculated above when assuming isothermal conditions.

Increasing the Thermal Efficiency of Torpedo Engines.—The power of a torpedo (compressed air) engine can be considerably increased by the addition of heat from some heating agent, or by burning some combustible substance and using the air charge to support combustion. The latter method supplies the more prolific means of augmenting the supply of energy, and is probably more capable of being safely used, because the rate of combustion can be more easily regulated to suit requirements.

With supplementary heating, the water usually surrounding the engine has an adverse effect on efficiency, because it takes away heat from the working substance instead of maintaining its temperature as in the simpler case of using compressed air only. This removal of heat can be only efficiently prevented by lagging the engine cylinders and other working parts exposed to the action of the cooling water, and might possibly be best attained in practice by excluding the water from the engine chamber, and by making the surrounding air serve as a non-conductor of heat from the engine to the water outside the torpedo.

Only a very rough comparison can be made, but it is known from petrol motor experiments that 105 lb. weight of air will support combustion for at least 7 lb. weight of petrol or spirit. In quantity the charge of, say, 105 lb. weight of air compressed to 2400 lb. per square inch pressure develops about 50 horse-power for 1 minute, representing the energy of

$$50 \times 42{\cdot}3 = 2115 \text{ b.t. units.}$$

The available heat energy of 1 lb. of petrol = 16,500 b.t. units about; and if only one-half of this energy can be conveyed to the

engine, there is still four times as much energy to be derived from it as can be derived from the air charge of the torpedo, when no heating is affected.

Steam Engines. Internal and External Work done during Evaporation.—To illustrate the principle of internal and external work, an example may be taken of a vertical cylinder containing 1 lb., by weight, of water on which a piston rests and exerts a *constant pressure* (Fig. 277). Let the pressure on the piston be represented by P, the corresponding pressure per square inch by p, and the area of the piston on which the pressure is exerted by A; then—

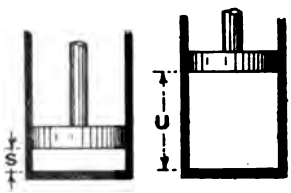


FIG. 277.—Work done by Steam.

$$P = p \,.\, A.$$

If all the water be evaporated under a pressure, p, then the piston will rise to a certain height; call this height u, in feet as measured from the bottom of the cylinder, and let s be the original depth of the water. Then the work done by the steam overcomes the resistance of the piston through a space, $u - s$, and therefore, during evaporation—

$$\text{External work done} = p \,.\, A \,.\, (u - s).$$

In this equation $A \,.\, s$ is the volume occupied by 1 lb. of water and is the specific volume of the water (*i.e.* it is equal to unity, 1), and therefore $A \,.\, s = 1$. Again, $A \,.\, u$ is the specific volume of the steam corresponding to a pressure, p, and is represented by v in the tables for saturated steam; so that—

$$\text{External work done} = p \,.\, (v - 1) \text{ in foot-lb.; or}$$
$$= \frac{p \,.\, (v - 1)}{780} \text{ in b.t. units.}$$

This external work grows with the increase of temperature, as found by experiment; but it grows very slowly, and the rate of augmentation diminishes with the increase of temperature. Equation is that for the saturation curve—

$$P \,.\, V^{1 \cdot 0645} = \text{constant.}$$

During the process of evaporation only the above quantity of

heat, $\frac{p.(v-1)}{780}$, can be utilised in doing work, and the remainder, a very much larger quantity, enters the mass of the water and is used up in overcoming internal molecular cohesions and molecular variations of form and shape, *i.e.* resistances which are invisible and only manifested by the results. Therefore it may be stated that—

(*a*) *Internal latent heat* is that necessary to overcome the internal resistance of the water opposed to its conversion into steam. And—

(*b*) *External latent heat* is that necessary to raise the piston, and which, therefore, does no work in the mass of the water but is used on external bodies.

The sum of these two quantities represents the latent heat L of formation; and denoting the internal latent heat by R, and expressing each part in b.t. units, then—

$$L = R + \frac{p.(v-1)}{780}.$$

If the total heat H of formation of steam is required, then—

$$H = S + L = S + R + \frac{p.(v-1)}{780}, \text{ where S = sensible heat.}$$

When $p = 315$, the value of v is about 88, and when $p = 14{\cdot}7$ (atmospheric pressure), $v = 1643$; therefore, approximately, $v = v - 1$, and is approximately the same at all ordinary pressures.

And $$H = S + R + \frac{p.v}{780}, \text{ approximately.}$$

If only a part, x, of the water is converted into saturated steam, and a fraction, $(1 - x)$, remains as water in suspension and mechanically mixed with the steam (as commonly found in practice), then—

$$H = S + x.R + \frac{x.p.v}{780},$$

and the ratio between the internal and external latent heat does not change.

From the above considerations, it is evident that when steam is generated under constant pressure—as in a boiler, after steam has been raised to the working pressure—that the measure of the external work done during evaporation is = the product of the

pressure and the change in volume (from water to steam at the working pressure); and adopting the proper units for saturated steam, this product approximately $= p \,.\, v =$ external work done during evaporation at constant pressure.

Any further increment of work obtained from the steam is done at the expense of the internal energy, contained in the steam in the form of heat, and in practice a relatively large quantity of work is so obtained by utilising the expansive quality of steam. In the formula for the mean pressure (and from which it is evident that the work done *during admission* is practically constant for the same weight of steam at all pressures), the expansion curve of pressures is assumed to follow the hyperbolic law; then—

$$Pm = \frac{p + p \,.\, \log_e r}{r};$$

and multiplying each side of this equation by v ($v = 1$, for 1 lb. by weight of steam or water)—

$$Pm \,.\, v = \frac{p \,.\, v}{r} + \frac{p \,.\, v \,.\, \log_e r}{r},$$

the increment is expressed by the last quantity. This increment increases with r, but in a rapidly decreasing ratio within the practicable limits of ordinary pressures and temperatures.

The action of steam expanding in a cylinder (Fig. 278) behind a piston may now be considered:—

1. When the piston is moved by some external force so that the steam (OD), *originally saturated*, expands and no external work is done, the quantity of heat energy contained in the steam remains unchanged if we neglect radiation, conduction, and other sources of loss. The pressure becomes less by expansion, as the steam has more space to fill, and the total heat necessary to maintain the steam in a

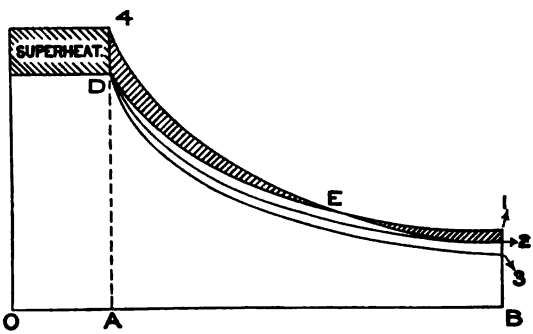


FIG. 278.—Expansion of Steam under Various Conditions.

state of saturation is reduced (see table, page 18); consequently a small quantity of heat is released from its duty in maintaining the state of saturation, and the steam becomes superheated (curve D1), *i.e.* it is raised to a temperature above that necessary to keep it in a state of saturation. The specific heat of superheated steam near the saturation temperature is about 0·48, and therefore for each b.t. unit set free per 1 lb. weight of steam the temperature of each 1 lb. will be raised 2·083° F. above the temperature of saturation; steam under these conditions being considered to act almost exactly as a gas.

2. When the piston is moved by some external force such that the steam (OD), *originally saturated*, expands and does some external work exactly equivalent to the heat set free by expansion, or that used to superheat it when no external work is done, then the quantity of heat remaining in the steam is just sufficient to keep the steam in a state of saturation (curve D2). The same condition may be obtained in practice—but is unusual—either when the steam, in addition to doing none or a small quantity of work, gives out heat by radiation, etc., or when the steam does work on the piston and receives heat from an external source, such as the jacket or other source, in sufficient quantity to maintain the steam in a state of saturation.

3. When the piston is not acted on by any external force and the steam (OD), *originally saturated*, expands and does work, as in the ordinary steam engine when unjacketed or jacketed, heat is abstracted from the steam and transferred into work done on the piston. In this case, generally the steam at the end of the expansion contains an insufficient quantity of heat to maintain it in a state of saturation; condensation therefore takes place and the pressure falls, in addition to the fall caused by expansion, and the steam becomes wet (curve D3). In the ordinary stage-expansion engine there is usually a drop in pressure when the steam enters the receiver between the stages, during which drop expansion takes place and no work is done (as in case 1 above), and some heat is therefore set free which either tends to re-evaporate any moisture which may be present in the steam, which is the usual condition, or tends to superheat the steam if already saturated, which is a very unusual condition.

4. When the piston is not acted on by any external force and the steam (O4), *originally superheated*, expands and does work on

the piston, as in steam engines using superheated steam, heat is abstracted from the steam and transferred into work done. The steam remains superheated (curve 41) at the end of the expansion if the work done, expressed in b.t. units, is not greater than $(T - t) . K_p$—

where T = Temperature of the superheated steam on supply to engine;

t = Temperature of same steam when in state of saturation;

K_p = Mean specific heat of superheated steam at constant pressure, between the temperatures T and t.

If the work done is only slightly in excess of this quantity, the steam at the end of the expansion (curve 42) may be in a state of saturation, *i.e.* when the work done, in b.t. units, $= (T - t) . K_p + (H - h)$, where—

H = Total heat of formation of steam at the initial pressure, and at the original temperature t;

h = Total heat of formation of steam at the final pressure produced by the expansion.

If the work done is in excess of the quantity $(T - t) . K_p + (H - h)$, then the steam becomes wet, as the quantity of heat contained in it at the end of the expansion is less than h and insufficient to keep it in a state of saturation.

In many three-stage expansion engines which may be supplied with steam considerably superheated, the steam remains superheated during the first stage (or H.P. cylinder) and a moderate amount of superheat is carried into the second stage (or M.P. cylinder); but during the expansion in the latter stage the steam passes from the superheated, through the saturated, into the wet state. In the third stage (or L.P. cylinder) the steam is generally wet throughout, notwithstanding partial re-evaporation, or tendency to superheat, obtained by the drop in pressure in the L.P. receiver and by any heat which may be received into the steam through the medium of the cylinder steam jacket.

The possible practical value of superheat in steam engines is much greater than might appear from the above considerations. In the first place, the maximum efficiency, considered as a perfect heat engine, is increased by the increase in temperature from T to T_s, because in place of an efficiency, $\frac{T - t}{T + 461}$, for saturated steam, an efficiency of $\frac{T_s - t}{T_s + 461}$ is obtained for superheated steam.

In the second place, superheat prevents initial condensation of the steam in one or more cylinders, and dry steam, especially when superheated, being a very poor conductor of heat, allows only a very small quantity of heat to escape through the medium of the cylinder walls. Steam when not superheated, or slightly moist, as it is usually when entering any cylinder, readily condenses from the fall in temperature caused by the comparatively cool cylinder; small globules of water are formed in the body of the steam, and, being deposited on the cylinder walls, rapidly convey heat away, thus occasioning a loss which does not occur when the steam remains superheated. As the cylinder condensation loss is less when superheat is used, a greater quantity of work is obtained from each 1 lb. weight of steam used in the engine. A greater quantity of heat is necessarily supplied to the steam, but the greater proportion of useful work obtained from superheated steam increases the efficiency by economising the heat supplied by the fuel.

Action of Steam Expanding in a Convergent-divergent Nozzle.—In "impulse" turbines of the De Laval type the steam is allowed to expand in a nozzle adiabatically (*i.e.* without either taking in heat from or giving heat out to external bodies); work is done by the steam in promoting motion in itself by reason of a fall in pressure and a corresponding fall in temperature. The greater the fall in pressure and temperature, the greater becomes the acceleration of velocity and energy of motion, or kinetic energy.

The nozzle may be any convenient length, either very short (as in the De Laval turbine, only 2 or 3 inches) or very long (as in turbines of the Parsons type, where the nozzle extends for the whole axial length of the blading). In the ideal nozzle, the area first converges to about 58 per cent of the original area; and then diverges to give the necessary increase in area at the discharge end. In calculating the discharge area, it should be clearly understood that work is being done by the steam inside the nozzle in creating motion of itself and that, therefore, heat is being converted into the kinetic energy of motion. If the steam be originally dry and saturated, some of it must necessarily be condensed to water during the expansion.

Fig. 346 represents a convergent-divergent nozzle, and the subject is dealt with in Chapter XXXIV.

PART VII

PROPULSION

CHAPTER XXIV

SHIP OR HULL RESISTANCE

WHEN a body of wedge-shaped form is completely submerged in a fluid, and moves through the fluid in the direction of its length, the

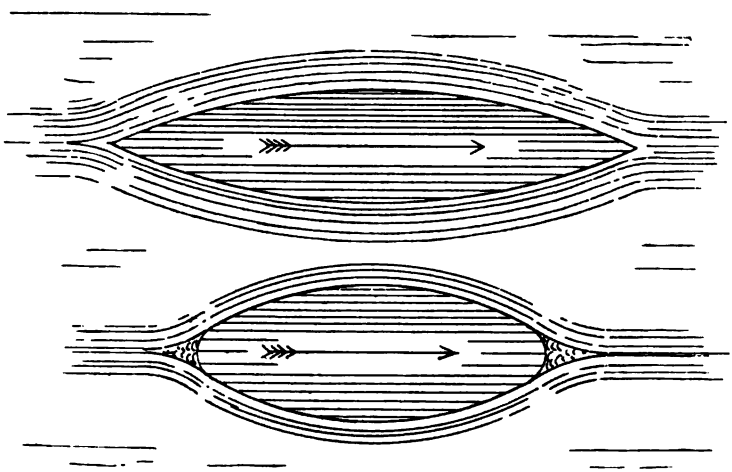


FIG. 279.—Stream Lines.

fluid opens out in front of the body, moves round it in definite lines, called *stream lines*, and closes in behind; the pressure which is exerted by the fluid on the front of the body is nearly balanced by that behind it. These stream lines are of perfectly definite form,

which differs according to the shape of the body. In all cases of ordinary fluids the body must be of nearly oval shape, and of gradual curvature for perfect stream-line motions. If sharp corners are presented to the fluid passing round them, some energy is dissipated, and the head resistance is consequently increased by the diminution of the pressure on the stern. In the actual case of a ship moving through the *water* a considerable resistance to motion is experienced. This is due to four causes:—

1. The fluid is imperfect, possessing the property of viscosity.
2. The surface over which the water passes is not frictionless.
3. Eddies are formed, because the body is not of perfect shape.
4. The body moves on and near the surface of the water and not entirely below it, and thus waves are formed.

1. **Viscosity of Water.**—When at rest, water is almost a perfect fluid, but when in motion it possesses the property of viscosity—that is, the friction among the particles of the fluid resists a change of form or distortion, but not a change in volume. A force producing such changes is proportional to the rate at which such changes take place.

The whole of the laws which are used for estimating the power required for the propelling machinery are based on experimental data, and only the frictional resistance of the under-water surface of the ship's sides (called the *wetted* surface) can be measured with any degree of accuracy. This measurement is obtained from model experiments or actual ships.

2. **Frictional Resistance.**—If a very thin plank is completely immersed and towed through the water, it is found that:—

(*a*) Frictional resistance F increases with a greater co-efficient, μ, of roughness of the surface; this is the resistance with which we have to deal in sea-going ships—

$$(F \propto \mu).$$

(*b*) Frictional resistance increases as about the *square*, actually the 1·84 power of the speed S or velocity—

$$(F \propto S^{1\cdot84}).$$

(*c*) Frictional resistance increases directly as the area A of the surface immersed—

$$(F \propto A).$$

(*d*) Frictional resistance increases with the length L of the surface immersed, and also with the depth of immersion; but the increase due to depth of immersion can be neglected, because it is of small importance with present types of ships—

$$(F \propto L).$$

From which, $$(F \propto A \,.\, L \,.\, \mu S^{1{\cdot}84}).$$

If the plank has 100 square feet of clean varnished surface, such as is found on a newly docked and painted ship, the power required to move it through the water at a speed per hour of—

(10 × 1), or 10 knots, is from 4 to 5 horse-power, or (4×1^3);
(10 × 2), or 20 knots, is from 32 to 40 horse-power, or (4×2^3);
(10 × 3), or 30 knots, is from 108 to 135 horse-power, or (4×3^3);

or, in other words, the horse-power required to overcome the frictional resistance of a newly docked and painted ship varies as about the *cube of the speed.*

As a ship goes through the water the rough surface of the sides causes a dragging action, which induces the water to flow at some speed which is, of course, less than that of the ship. This induced flow forms the medium in which the propeller works, and thus the propeller acts on water which is already set in motion. The speed of the water thus set in motion by the frictional wake is found to be something below 10 per cent of the speed of the ship.

3. **Eddy-making.**—Eddies are caused by the varying velocities set up by the dragging action of the ship. The particles roll over each other, and, accumulating varying velocities and energy, react on each other. All this disturbing action is a waste of the propelling force. In the same manner—that is to say, both by lack of fairness of form and by surface friction—the action of the propeller blades sets up another series of eddies, due to the blades acting obliquely to the direction of motion and to the frictional drag of the surfaces of the blades.

In well-formed ships, at low speeds, the total eddy-making resistance is small—not exceeding 10 per cent of the total resistance, and is sometimes less. Eddying resistance is principally due to the defective form of entrance and run, particularly the latter, because for structural reasons the stern lines cannot usually be made quite as fine as those at the bow.

4. **Wave-making.**—Wave-making may be divided into two

principal parts—the *divergent* waves thrown off by the ship from the bow and sides, and the wave which follows the ship, called the *stern* wave.

The divergent waves, leaving the ship, culminate in one or more waves of large dimensions, and produce no corresponding reaction or forward pressure under the stern; and consequently their creation is a loss of propelling power.

The following or stern wave is not an entire loss of power, because part of it reacts on the stern, and thus assists the ship by its forward velocity or thrust; and the energy expended is partially returned in this way. It may, however, be considered that disturbance created by the passage of the ship through the water must result in some waste of the propelling power. The resistance due to wave-making depends on the fineness of entrance and run, especially the latter, combined with their lengths in relation to the immersed length of the vessel. Wave-making is comparatively unimportant at low and moderate speeds, but it absorbs a large proportion of the power, probably 15 to 20 per cent, at the highest speeds.

Augment of Resistance.—In screw-propelled ships, the ship or hull resistance is increased or augmented by the action of the screw in diminishing the pressure of the water under the counter or stern end of the vessel. This resistance is considerably augmented by any increase in speed; and at a certain high speed, which is peculiar to every individual vessel, any further increase in speed becomes impossible. In some cases the augment of resistance has been confounded with that caused by the defective form of screw employed, which actually limited the speed, and not the form of the hull to which resistance augmentation is properly due.

In the trials of the *Medea* class an addition of 3000 I.H.P. produced only a doubtful increase of about one-tenth of a knot in speed; subsequently in the *Latona* class, in all respects similar except an increased length of 35 feet and a consequent increased displacement of 800 tons, a higher speed was obtained without the additional 3000 I.H.P.

"Mauretania" Model Experiments.—Probably the most interesting and important experiments made and published in recent years were made by Messrs. Swan, Hunter, and Wigham-Richardson, and the Wallsend Slipway and Engineering Company,

with a model of one-sixteenth the actual dimensions of the *Mauretania*, and about 47½ feet in length. The trials were carried out in a dock on the Tyne, and events have proved their great accuracy in actual practice. The motive power of the model was supplied by electric motors driving four screws and shafts, for which power was supplied by accumulators stowed in the launch itself. Considerable difficulty was at first experienced from rapid fouling of the wetted surface, but eventually this element of inaccuracy was overcome. Incidentally it may be mentioned that, after seven weeks, the foulness of the bottom caused an increase of 25 per cent in power at full speed, and differences quite as large obtained after only two weeks when the fouling was not interrupted by speed trials. Dynamometers were carefully arranged for taking readings under various conditions, and the instruments were tested from time to time by means of a special form of Prony brake. Several hundreds of trial runs were made, and the actual pitch and diameter and other particulars of the screws obtained. There appeared to be no material difference by using either twin or quadruple screws, and eventually four screws were adopted, the outward pair being placed 78 ft. 11 in. before the middle pair; in this position the forward propellers did not appear to interfere with the efficiency of the after pair.

Influence of Depth of Water on Speed.—For many years past it has been known that shallow water affected the speed which might otherwise be obtained for the same vessel. The difference in resistance (it is now known that it may be either increased or decreased) is due to wave-making; and the form of wave created is now generally termed the "canal wave." Investigations by Messrs. W. W. Marriner and H. E. Yarrow, by Messrs. Thornycroft, and by Colonel Giuseppe Rota of the Italian Navy, have been supplemented and summarised by Mr. Sydney Barnaby in the Watt Anniversary Lecture at Greenock, January 19, 1906. (See *Engineering*, February 2, 1906.)

These results are somewhat curious and appear to be applicable to all high-speed vessels. Roughly, for comparative purposes of various classes of vessel, the depth of water should be 10 to 11 times the load draught of the vessel, when of fine lines with a length of about 9 or 10 times the beam. In shallower water, the speed gradually decreases for full power until the depth is only 5½ to 6 times the draught of water, which is about the most unfavourable

condition for full power. With a further decrease of depth, the speed gradually increases until it becomes a maximum at a depth of about twice the load draught. But at half-power the drag of the stern wave, due probably to its greater wave length, decreases the speed very much below that obtainable for the same power in deeper water.

With ships of fine lines, but of a length of only 6 to 7 times the beam, such as large battleships and cruisers, the problem becomes more complicated. To be on the safe side, it is probably in water of at least 11 times the draught that the best speed will be obtained at all powers. The *Mauretania* model experiments were run in a depth of 24 feet, the draft of the model being $24\frac{3}{4}$ inches.

INFLUENCE OF DEPTH OF WATER ON SPEED

Depth of Water.	Power or Speed.	Remarks on Effects.
27-knot German Torpedo Boat, 200 ft. long, 347 tons displacement.		
Various	12 knots	Inappreciable
130 ft. and upwards	12 to 17 knots	Insignificant
23 ft.	Half power	5·64 knots *less* than in 200 ft. depth
23 ft.	Full power	0·84 knots *more* than in 200 ft. depth
80 ft.	27 knots	Most unfavourable
25½-knot Torpedo Boat Destroyers (River Class), 225 ft. long, 600 tons.		
25 to 35 ft.	Full power	Highest speed obtained
120 ft. and upwards	,,	Most favourable
50 ft.	,,	2 knots *less* than in 100 ft. depth
240 ft.	,,	1 knot *more* than in 100 ft. depth
60 to 70 ft.	,,	Most unfavourable
26-knot Torpedo Boats or Coastal Destroyers, 168 ft. long, 230 tons.		
20 ft.	Full power	Most favourable
33 ft. (?)	,,	Most unfavourable
98 ft. and upwards	,,	No appreciable increase
33-knot Sea-going Destroyers, 260 ft. long, 800 tons.		
32 ft.	Full power	Most favourable
50 ft. (?)	,,	Most unfavourable
150 ft. and upwards	,,	No appreciable increase
36-knot Sea-going Destroyer, 345 ft. long, 1800 tons.		
36 ft.	Full power	Most favourable
55 ft. (?)	,,	Most unfavourable
170 ft. and upwards	,,	No appreciable increase

Effective Horse-Power and Thrust.—The I.H.P. is a measure of the work imparted by the steam to the pistons, and part of it is used in overcoming the frictional resistances of the mechanism, at

the thrust block, and at the propeller itself, and also in creating wasteful disturbance of the water. Consequently, only 50 to 60 per cent of the I.H.P. is actually used in promoting *effective thrust*, and this proportion of the power is called *effective horse-power, or E.H.P.*

If a ship is towed through the water the effective thrust is approximately equal to the pull on the tow-rope; but it is not an exact measure of the pull, because it depends on the various efficiencies involved. For warships of ordinary form and shape the tow-rope resistance is generally equal to about 90 per cent of the effective thrust, and the E.H.P. may be taken as equal to the thrust H.P., or about one-half the I.H.P.

For ordinary purposes of calculation and comparison, it is more convenient to use a figure calculated directly from the I.H.P., called the *indicated thrust*, and which is generally expressed in pounds. Thus—

$$\left.\begin{matrix}\text{Indicated thrust,}\\ \text{in pounds}\end{matrix}\right\} = \frac{\text{foot-lb. of work done per minute}}{\text{distance through which resistance is overcome}}$$

$$= \frac{\text{I.H.P.} \times 33{,}000 \text{ (per minute)}}{\text{pitch} \times \text{revolutions of screw (per minute)}}$$

$$= \frac{2 \cdot P \cdot L \cdot A \cdot N}{\text{pitch} \times N}$$

$$= \frac{2 \cdot P \cdot L \cdot A}{\text{pitch}}.$$

From which—

$$\left.\begin{matrix}\text{Effective thrust,}\\ \text{in pounds}\end{matrix}\right\} = \frac{2 \cdot P \cdot L \cdot A}{\text{pitch}} \times (50 \text{ to } 60 \text{ per cent});$$

or—

$$\left.\begin{matrix}\text{Pull on the tow-}\\ \text{rope, in pounds}\end{matrix}\right\} = \frac{2 \cdot P \cdot L \cdot A}{\text{pitch}} \times (45 \text{ to } 54 \text{ per cent}).$$

The effective pull exerted by one ship in towing another is therefore independent of the number of revolutions of the towing ship's propeller. When the towing ship is moving at a greater speed than the ship towed, there is an increased difference of momentum, which momentarily increases the pull, and generally breaks the tow-rope; this is not an effective pull.

For towing purposes, the propeller should be proportioned to work effectively and without undue slip, and a large blade area is necessary to take the thrust. Apart from the important consideration of tonnage displacement, the most effective towing ship is one which requires the least number of revolutions of the propeller to travel one knot; that is, a propeller with large pitch and blade area. For this reason, as well as the small area of the thrust collar surface, *ships propelled by Parsons' turbines are not suited to towing purposes*; but with thrust blocks properly designed and enlarged for the purpose, turbine propelled vessels might be made fairly effective towing ships at moderately low speeds.

A more convenient form of the above equation is—

$$\text{Pull on the tow-rope, in lb.} = \frac{\text{I.H.P.} \times 33{,}000}{\text{N} \times \text{pitch}} \times \frac{45 \text{ to } 54}{100}.$$

In any man-of-war, or other vessel not primarily constructed for towing purposes, only about one-half the I.H.P. is available for towing any vessel of about the same tonnage, because the full number of revolutions cannot be obtained with the extra load, and therefore the greatest pull is probably about

$$= \tfrac{1}{2} \times \frac{\text{I.H.P.} \times 33{,}000}{\text{N} \times \text{pitch}} \times \tfrac{1}{2} \text{ about.}$$

EXAMPLE.—For vessels of the *King Edward* class, of 18,000 I.H.P., pitch screws about $16\frac{1}{2}$ feet, and N = 90 r.p.m. at about half-power—

$$\text{Pull on the tow-rope} = \tfrac{1}{2} \times \frac{18{,}000 \times 33{,}000}{90 \times 16\frac{1}{2}} \times \tfrac{1}{2}$$

$$= 100{,}000 \text{ lb.}$$

$$= 44{\cdot}6 \text{ tons about.}$$

In the above example, if it be assumed that the full I.H.P. can be developed when towing, with the r.p.m. of about 120: then the theoretical maximum—

$$\text{Pull on tow-rope} = 150{,}000 \text{ lb.} = 67 \text{ tons, about.}$$

Air Resistance.—The resistance due to the air is becoming of increasing importance, because both the surface exposed to air and the speed are fast increasing. In destroyers, careful design has probably reduced air resistance; but the precautions taken by contractors to remove or minimise all possible obstructions show

the importance attached to it. A destroyer steaming against the wind causes an indraught abaft the after funnel or other obstruction, similar to the windage of a railway train.

Mr. Froude's experiments showed that at a velocity of one foot-second the air resistance on a plane area is 0·0017 lb. per square foot. Assuming that the athwartship area of a 30-knot destroyer, which is exposed to the air, is about 200 square feet, the velocity 50 foot-seconds (or 29·5 knots), and that the air resistance varies as the square of the velocity; then—

$$\text{Total air resistance, in pounds} = (200 \times 0{\cdot}0017) \times 50^2$$
$$= 2 \times 17 \times 25 = 850 \text{ lb.}$$

If instead of steaming through still air, as above, the destroyer is steaming against a 15-knot breeze, the resistance is increased to about 2000 lb., and the I.H.P. required to overcome this resistance is—

$$R \times S = \frac{2000 \times 75 \times 60}{33{,}000} = 275 \text{ nearly.}$$

About 6000 I.H.P. is required to propel this destroyer at 30 knots, and the percentage of this expended in overcoming the air resistance when steaming against a 15-knot breeze—

$$= \frac{275 \times 100}{6000} = 4{\cdot}6 \text{ per cent.}$$

In the *Mauretania* model experiments it was demonstrated that 12 per cent more power would be required to propel the vessel at 25 knots against a 25-knot wind than when moving through still air.

Influence of Weather.—Throughout these remarks it has been assumed that the vessel is propelled through still water. In practice, of course, there is a large difference, due to the roughness of the sea, produced by the wind alone, or wind and current. Sea speed is best maintained with great length, which generally involves increased displacement, and even when both these are incorporated the influence of form or shape may neutralise the advantages which might be otherwise expected. A 20 per cent increase in power, and a decrease of 20 per cent in speed, is not unusual when steaming against a head sea, and the difference is principally due to the pitching motion. If the ship can be turned

about so as to take up a rolling motion when running before the sea, the speed may be increased, and the power decreased to normal. Experiments in smooth water, on which nearly all theoretical investigations are based, are not always suitable to sea-going purposes. Results should be tabulated of the experience gained under sea-going conditions, and these should be shown on a large-scale curve for further reference. At present there is practically no sea-going data on which to base any reliable calculation for future designs, although such details can be obtained in everyday experience.

Power and Speed.—Within the ordinary limits of power (between one-fifth and three-fourths that of full power) and speed, only the frictional resistance need be considered. It varies as about the 1·84 power of the speed as found by experiment; but for ordinary purposes the *square* is, perhaps, a closer approximation.

First law: *The resistance varies as the square of speed* ($R \propto S^2$).

Let S = speed in feet per minute, and let R = resistance in pounds at this speed. (R is the actual pull on a tow-rope required to move the vessel at a speed S.)

$$\text{Then } R = K \times S^2$$

where K is a constant.

Multiply both sides of this equation by S; then—

$$R \times S = K \times S^3.$$

But R × S = work done in overcoming a resistance R through a space S, and, if expressed in foot-pounds and divided by 33,000, is equal to the I.H.P. required to overcome the resistance.

Second law: *The I.H.P. varies as the cube of the speed* ($\text{I.H.P.} \propto S^3$).

Fuel, Speed, and Power.—Within moderate limits the consumption of fuel per hour per I.H.P is the same at any power, and—

$$\frac{\text{Consumption of fuel per hour}}{\text{Speed in knots (per hour)}} \propto \frac{\text{I.H.P.}}{\text{speed}} \propto \frac{S^3}{\text{speed}} \propto S^2.$$

Therefore—

Third law: *The consumption of fuel (F) per mile varies as the square of the speed,* or $F \propto S^2$.

EXAMPLE.—If 40 tons of coal are burnt at a speed of 20 knots, how much would be burnt at 21 knots?

Evidently the consumption (f) per mile = 2 tons, and

$$\frac{F}{f} = \frac{S^2}{s^2}.$$

Therefore $$F = \frac{S^2}{s^2} \times f = 2{\cdot}2 \text{ tons per mile}$$

$$= 46{\cdot}2 \text{ tons per hour.}$$

Fourth law: *The total fuel consumption* (F) *for any distance* (D) *varies as the square of the speed multiplied by the distance,* or $F.D \propto S^2D$. This is evidently the same as the third law, each side being multiplied by the distance D.

EXAMPLE.—If 200 tons of coal are consumed in steaming 800 nautical miles at 11 knots, how much will be required for a distance of 1210 nautical miles at 10 knots—excluding the coal used for auxiliary purposes in each case?

Total consumption, $F.D = 200$ tons

and $$\frac{f.d}{F.D} = \frac{s^2.d}{S^2.D}.$$

From which, $$f.d = 200 \times \frac{100 \times 1210}{121 \times 800} = 250 \text{ tons.}$$

The steam trials of the Japanese battleship *Kashima* are a fairly recent demonstration of the value of n in the theoretical equation, I.H.P. $\propto S^n$, which show that at about full power the I.H.P. $\propto S^{3{\cdot}16}$, and at about one-third power the I.H.P. varies as $S^{2{\cdot}87}$, both values being based on the correctness of the observations at a power of about one-sixth the full power.

H.I.J.M.S. *KASHIMA*

Revs. per Min.	I.H.P.	Speed in Knots.	Value of n, the Index of S^n.
69·5	3,030	11·136	...
89	6,275	14·27	2·87
102	9,160	16·323	2·87
110	11,400	17·204	3·02
113·6	13,000	18·000	3·01
123	17,280	19·242	3·16

Comparisons of several naval vessels, of different types, show the same practical tendency, but unfortunately the general inaccuracy of the speed, *by log,* at one-fifth power renders in this connection nearly the whole of these comparisons valueless. As an example,

the *New Zealand* at 16·9 and 18·6 knots has index values of 1·9 and 2·11 respectively; but if a comparison is made between these two speeds and the respective powers, independently of the logged speed of 9 knots (obviously inaccurate) at one-fifth power, the index value is 3·6, which is probably more nearly correct.

Economical Speed.—The greatest distance which a ship can steam on a consumption of 1 ton of coal is the most economical speed; but for any vessel this distance is practically indeterminate because of the variation of wind and sea conditions, the cleanliness or otherwise of the ship's bottom, the quality of the coal; the consumption of fuel and steam for auxiliary purposes, and other minor reasons. Generally for large and older battleships and cruisers (whose I.H.P. is nearly equal to the tonnage displacement) the economical speed is at about one-fifth to one-sixth power. In practice it is necessary to construct a simple tabular statement—including in it the coal used for auxiliary purposes—of the coal consumption at each knot speed between the lowest obtainable and the full speed, and to calculate the distance steamed per ton of coal at each of these speeds.

A rough estimate of the loss from the state of the ship's wetted surface is 1 per cent of the full speed for each month since the ship was last undocked; this is fairly accurate up to a period of six months, but after this the loss is only very slightly increased.

In Fig. 280, curves have been drawn to show the coal burnt per hour in tons; the coal consumption per I.H.P. per hour in lb.; and the distance run per ton of coal burnt relative to the speed in knots, for one of the 30,000 I.H.P. cruisers. The coal used for auxiliary purposes is included in the consumption per hour in tons and in the distance run, but not in the consumption per I.H.P.-hour. From the curve it appears that the economical speed is sharply defined at E, for 10 knots; but this is only the case when the conditions are identical and in exact accordance with the conditions thus represented.

Taking all the above remarks into consideration, it may be stated that the economical speed for any vessel is probably about two-fifths of the full speed, obtainable with full power, corresponding to the state of the wetted surface. This speed under ordinary conditions in recently constructed vessels would correspond to a power somewhere between one-fourteenth and one-twelfth the full I.H.P. (The difference between these and the older vessels consists

in the higher powers fitted and in the percentage required to overcome the engine friction.)

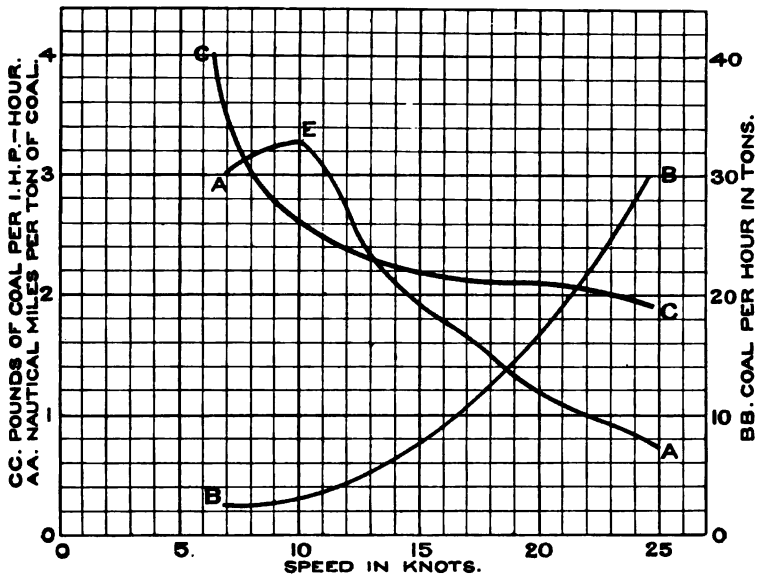


FIG. 280.—Curve of Performance.

Effect of Sea-going Conditions on Power and Speed.—In the tabular statement below, the effect of various changes of steaming conditions is very fairly shown, but in some of the results the speed does not appear to have been very accurately logged. The revolutions at very high speeds generally give a more nearly accurate measurement. The effect of greater depth of water is shown by a comparison of the contractor's trials off Chesil Beach (depth 17 to 20 fathoms) in September 1902, and the 8 hours' trial between Vigo and the Eddystone (depth 70 to 200 fathoms) on April 29, 1907. The conditions are almost identical except for the greater depth of water, which gave an increased speed of 0·4 knot.

The effect of increased frictional resistance of the wetted surface is shown by comparison of trials of September 24, 1902, and December 12, 1907; there is a slight gain in speed due to increase of depth of water, but nearly 2000 I.H.P. more are required to obtain the same number of revolutions. In February 1907, owing to foulness of wetted surface, the rate of revolution could not

be increased above 121 per minute although the engines were supplied with all the steam which they could utilise; the speed was logged at only 23 knots, but the revolutions should have produced 24·2 knots with a clean bottom. Under almost identical conditions, but with a clean bottom, a speed of 24·5 knots was obtained in April 1907.

A careful study of this statement together with the coal consumption on actual service at moderate speeds, shows that the period elapsing between the docking of large battleships and cruisers should not generally exceed six months. Docking for purposes of cleaning should be considered in conjunction with the coal and water bills. Apart from considerations of pure economy, frequent docking is a necessity for war purposes; only a small number of docks is available, and on the brink of war it would be impossible to dock all the ships at once. A loss of 1 to 1½ knots might also be attended with dire consequences.

Curves of Power and Speed.—By combining the speed and power, a curve can be constructed to show the power required for any speed within the limits of the curve, and for a short distance above and below these limits.

In Fig. 281 the horizontal line shows the speed in knots, and the vertical lines the powers required for the various speeds of a destroyer. The curve is drawn through the various points obtained from results of trials. By very carefully plotting, on a large-scale curve, the lower powers and speeds, and then continuing the curve until it cuts the vertical zero ordinate, it is found that some power is apparently indicated (as shown by the curve) when no motion is imparted; the power thus registered is that required to overcome the frictional resistance of the engines. This frictional

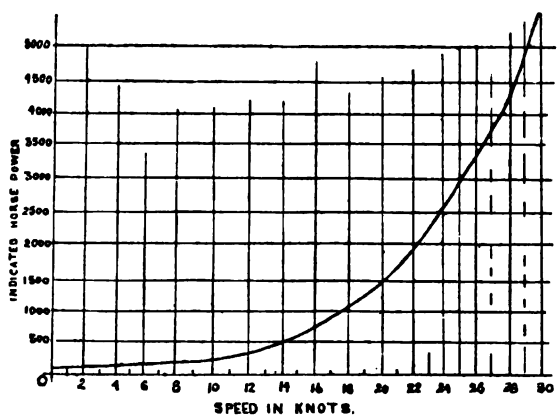


FIG. 281.—Speed and I.H.P.

H.M.S. *DRAKE*

Date of Trial.	Displacement in Tons.	Direction and Force of Wind.	State of Sea.	Time since last Undocked.	Duration of Trial.	I.H.P. Average.	I.H.P. per 1 Ton of Coal per Hour.	Revolutions per Minute.	Logged Speed in Knots.
				Months.	Hours.				
June 1902 .	14,216	Various, 2	Moderate to slight	1	8	30,849	1193	116	23·05
Sept. 23, 1902.	14,158	Various, 1 to 2	Smooth	¼	4 runs off Chesil Beach	31,454	...	122·4	24·11
June 1902 .	14,348	Various, 2	Moderate to slight	1	30	23,004	1240	105·4	22·08
Sept. 24, 1902.	14,100	Various, 1 to 2	Smooth	¼	4 runs off Chesil Beach	22,534	...	111·5	22·16
The above were contractors' trials, with original and new propellers.									
June 29, 1903 .	13,955	Ahead, 5	Slight swell	6	3	28,671	1125	119·23	23·0
Feb. 27, 1904 .	13,259	Beam	Head swell	4½	...	24,554	929	115·35	23·5
July 5, 1904 .	14,941	Bow, 1 to 2	Smooth	1	8	28,184	986	116·2	22·8
Jan. 5, 1905 .	14,941	Bow, 2	,,	2	8	29,690	1004	120·08	24·16
June 23, 1905 .	14,187	Bow, 1	,,	1	8	29,201	953	119·615	23·72
Oct. 31, 1905 .	14,941	Bow, 3	,,	5	8	31,061	969	121·4	24·25
Mar. 1, 1906 .	14,100	Various, 2	,,	9	8	28,632	957	117·6	23·5
June 29, 1906 .	15,144	Ahead, 2 to 5	Moderate	3½	7	Unreliable	...	107·0	20·4
June 15, 1906 .	14,738	Various, 2 to 3	Moderate to slight	3¼	Manœuvres 1	32,200	949	123·0	24·8
Aug. 25, 1906 .	14,448	Beam, 3 to 6	Rough	5½	Manœuvres 8	31,338	1036	120·0	23·2
Dec. 12, 1906 .	14,883	Various, 2 to 6	Smooth to heavy ocean swell	9	30	24,459	990	111·3	22·3
Feb. 1907 .	14,421	Various, 2 to 3	Slight swell	11½	1½ Manœuvres	30,006	1058	121·0	23·0
April 29, 1907 .	14,593	Bow, 1	Smooth	1½	8	31,717	1016	122·5	24·5

resistance is about 5 per cent of the full power, when the engines are running at full speed and with the propellers removed, and with the engines in good working order. From time to time it is a useful practice to plot the curve and compare it with a known good result. If there is any increase, it points to some excessive friction, and that the engines are absorbing undue power from inefficient adjustment of the mechanism.

In Fig. 282 a curve is shown of the I.H.P., revolutions, and slip of the (original) propellers, at various ascertained speeds, for

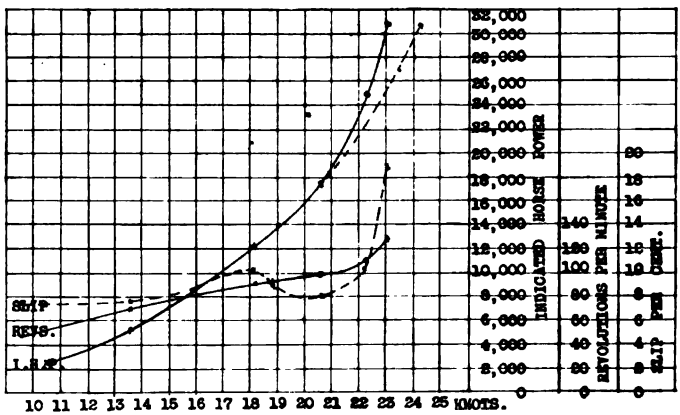


FIG. 282.—I.H.P., Revolutions, and Slip.

H.M.S. *Good Hope*; and also a part of a curve of performance as obtained by her sister ship, H.M.S. *Drake*, with improved propellers, for purposes of comparison.

It is noticed that the I.H.P. of the *Good Hope* increased rapidly at a speed of 22 knots, and the last knot required nearly 9000 I.H.P. for its attainment, thus emphasising the inefficiency of the propellers at the higher speed. For a similar increase of power the *Drake* obtained an increase of about 2 knots; and after changing the propellers of the *Good Hope*, about the same result was also obtained.

Curves of Coal Consumption.—In Fig. 283, the vertical ordinates represent the consumption of coal at the various speeds shown by the horizontal ordinates for H.M.S. *Amethyst* and *Topaze* on their trials. The enormous cost of speed is shown by the curve.

Willans Line.—The I.H.P. varies very nearly as the coal consumption, *i.e.* the coal consumption *per horse-power* is nearly constant

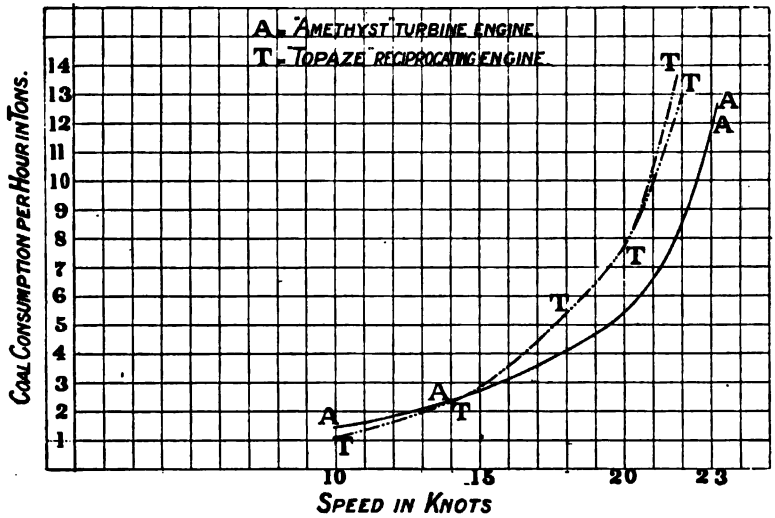


Fig. 283.—Coal Consumption and Speed.

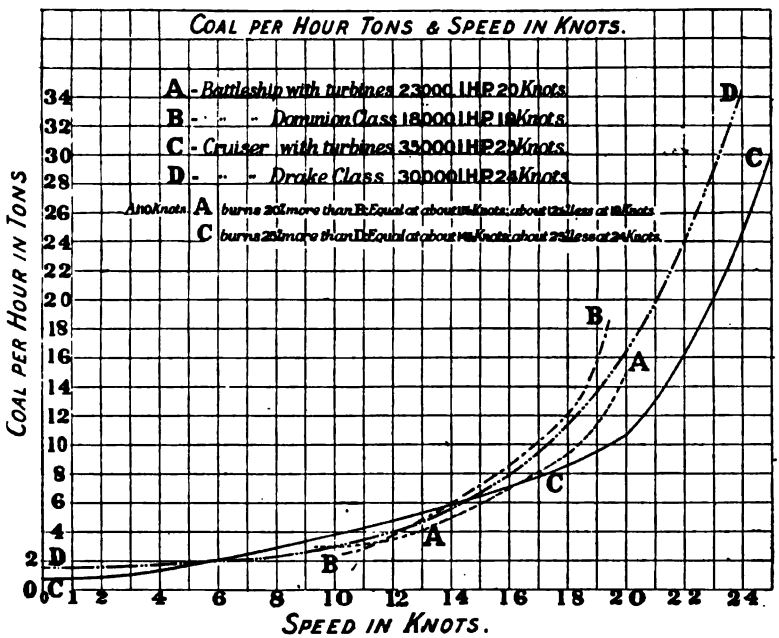

Coal per Hour Tons & Speed in Knots.
A - Battleship with turbines 23000 I.H.P. 20 Knots
B - " " Dominion Class 18000 I.H.P. 18 Knots
C - Cruiser with turbines 35000 I.H.P. 25 Knots
D - " " Drake Class 30000 I.H.P. 24 Knots
At 0 Knots A burns 20% more than B. Equal at about 18 Knots: about 1½ less at 16 Knots
C burns 25% more than D. Equal at about 19 Knots: about 2½ less at 24 Knots.
Coal per Hour in Tons
Speed in Knots.

Fig. 284.—Curves of Comparison, Turbine and Reciprocating Engines.

at all speeds. From some investigations by Mr. Willans, the curve showing the relation between fuel consumption and horse-power is called the *Willans line*, which for a small high-speed engine he found to be nearly *straight*. With large engines, as shown by the curve in Fig. 280, this is by no means the case in practice.

Fig. 284 shows curves of the coal consumed per hour, actual and estimated, for comparison of the relative advantage of using turbine instead of reciprocating engines. Another comparison, illustrated in Fig. 364A, shows the further advantage obtained by utilising some of the energy contained in the auxiliary exhaust.

Curve of Indicated Thrust.—It is convenient to express the results of trials in the form of a curve, from which the tendency to any abnormal excess may be readily detected. The efficiency of the propeller and of the mechanism should be fairly constant at all speeds and powers, within the limits for which they are designed, so that, instead of calculating the effective thrust H.P. for each speed, and then plotting the results on a curve, it is more convenient to simply plot a curve of the indicated thrust itself. The horizontal components of such a curve show the speeds in knots to scale, and the vertical components show the corresponding thrusts indicated in *pounds*.

Propulsion.—A vessel is caused to move through the water and over its surface by the propellers moving the water in such a way that motion of the vessel is produced in the required direction. The resistance must be overcome to produce motion, and the displacement of the water is a measure of the resistance. "Action and reaction are equal and opposite," and if all the resisting forces produced by the effort of propulsion be resolved into two parts, one of which is in line with the direction of motion of the vessel, and the other at any angle with this line, then the first may be considered as a measure of the work usefully expended, and the remainder as work expended in creating wasteful disturbance. From which it may be considered that the effective thrust is equal to the momentum of the stream driven directly astern.

For paddle-wheel propulsion, the area of the stream propelled sternward is evidently equal to the area of the columns acted on by the floats on each side of the ship; and if the floats are totally immersed, it is approximately equal to the area of two floats, one on each side of the vessel.

For screw propulsion, the area of the stream is approximately

equal to the area of a circle, whose diameter is that of the screw *less* an area due to the greatest area of the boss of the screw, which has no propelling effect. The momentum of the stream is equal to its mass multiplied by its velocity. Call this momentum M.V., and let A be the area of the column of water moving directly sternward.

The velocity of the stream is approximately equal to the velocity of the propeller through the water. And the action of the propeller tends to equalise any difference in these velocities.

And if p be the pitch of the propeller or propellers in feet, n be the number of revolutions of propeller per second, then the volume of water moving directly sternward per second in cubic feet is equal to $p \,.\, n \,.\, A$ in cubic feet.

The weight of one cubic foot of sea-water is 64 lb., and therefore the weight of water moving directly sternward per second $= 64 \,.\, p \,.\, n \,.\, A.$

But $M = \frac{W}{g}$, where W is the weight of water as above, and g is equal to 32, approximately.

$$\text{Therefore } M = 2 \,.\, A \,.\, p \,.\, n.$$

Let s = the speed of the ship in foot-seconds.

If the propeller and the ship are moving through the water at the same speed, the propeller simply cuts through the water without imparting any thrust. The power of the propeller to impart thrust depends on the difference between the speed of the propeller and the speed of the ship which carries the propeller, and therefore the velocity of the stream driven sternward by the propeller may be considered as $= (p \,.\, n - s)$, which = the velocity, V, of the mass, M, acted on per second.

Therefore the effective thrust = momentum of the stream

$$= M.V.$$
$$= 2 \,.\, A \,.\, p \,.\, n \,.\, (p \,.\, n - s) \text{ in pounds.}$$

As the area A cannot be accurately determined, this expression for the effective thrust is only an approximation; but it is useful for solving some of the problems presented by variations of results obtained in practice, particularly in the case of apparent negative slip.

CHAPTER XXV

THE PROPELLER—PADDLE, JET, SCREW, AND TURBINE-SCREW

Paddle Wheels.—In the early steamships paddles were used for propelling the ships through the water, and under good conditions of wind, weather, and proper immersion they still give a fairly high efficiency in comparison with the screw propeller.

The *radial paddle wheel* is the simplest to construct, as the floats are secured directly to the arms radiating from the centre of the wheel. In this case the only time when the float acts perpendicularly to the water, the surface of which is always considered horizontal, is when the radius arm is vertically below the centre; but at all other times the thrust is oblique, and consequently less efficient.

The *feathering paddle wheel* was introduced to overcome the disadvantages arising from this oblique action. A diagram is shown, in Fig. 285, of the usual method of feathering. A is the centre of the wheel, which is generally mounted on the projecting ends of the crank shaft D. B is another centre, horizontally in front of the shaft centre, and fixed to the outboard part E of the paddle box F, but inside the box itself. BK is a radius bar, and KC is connected with it at K by a pin-joint. KC is a lever, with its fulcrum at L, which is a point on the wheel circle; usually L very nearly coincides with C. The shaft revolves the wheel, which carries the floats with it through the concentric medium of the pin-joints L. The radius arm BK is constrained to work in a circle round B as centre, and at the same time L is constrained to work in a circle round A as centre. These two constraints produce an alteration in the movement of the floats (as shown in the figure), and at their entry to and exit from the water they are more nearly vertical, thus producing a more direct thrust.

The disadvantages of the paddle wheel compared with the screw are—

1. Greater variation of effect when the ship is rolling or

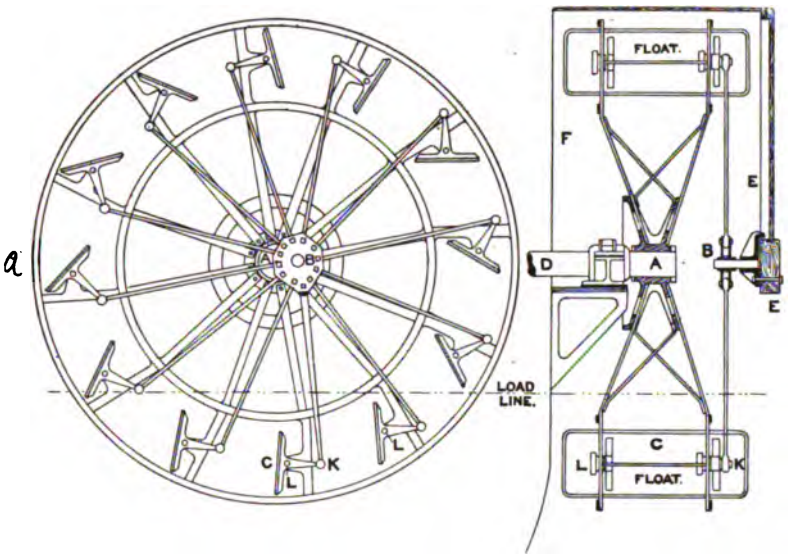


FIG. 285.—Feathering Paddle Wheel.

pitching; the floats are sometimes immersed too much and at other times too little.

2. The draught of water must be accurately maintained, or the floats will, if the ship be sufficiently weighted, be immersed to such an extent that propulsion will be difficult or inefficient. If weight be taken out, the floats will not get sufficient hold of the water. On long voyages, the coal burnt during the passage alters the draught of water and the immersion of the floats considerably.

3. The paddle wheel is always exposed to shot and shell, and it is therefore of comparatively little use for warships.

Jet Propulsion.—Various attempts have been made to propel ships by pumping water through pipes. The water is drawn from the sea and ejected with a greater velocity, and the momentum produced by this acceleration constitutes the impelling force. The frictional resistances of the channels through which the water passes absorb a large share of the power, and in addition the changes in direction considerably reduce the momentum of the ejected water.

Two similar boats, built by Messrs. Thornycroft many years ago, one propelled by a screw and the other by jet propulsion, with a horse-power of about 170 each, obtained $17\frac{1}{4}$ knots and $12\frac{1}{2}$ knots respectively; in other words, the screw-propelled boat could attain about the *same* speed for about *half* the power.

The screw in some ships, where partial tunnels are adopted to give a better run of the water to the propeller, is to some extent an extreme example of a jet propeller. If the tunnels are complete, as in the Thornycroft turbine arrangement used for shallow-draught boats (Fig. 289), the propeller simply performs the function of a helical pump with as little change in direction of flow of the water as possible.

Screw Propeller.—This is the usual method employed for propelling sea-going ships. The screw is placed under the counter or after-part of the ship, and is generally well immersed, so that it is little affected by the disadvantages which render the paddle inefficient; and, in addition, the propelling engines can be kept below the water-line and, therefore, for warships, under a fair protection from shot and shell.

The screw consists of a boss, fixed on the tail end of a fore-and-aft shaft revolved by the engine, which carries a set of blades radiating from it, and these, being placed obliquely, form parts of the threads of an actual screw. When revolved, the screw advances through the water just as an ordinary screw advances through wood.

If an ordinary screw, such as is used by carpenters, be driven into a board by a turn-screw, the distance it advances into the wood in one complete revolution is one *pitch*. The marine propeller is simply a very short length of an actual screw (Fig. 286), but of very coarse pitch, and the thread of which is cut very deep, equal to about two-thirds the radius of the outside diameter. The ordinary screw will produce only one blade, as there is only one *continuous* thread; but if there are two threads, which lie parallel, side by side, and a short length be cut off, there are two blades, thus producing a two-bladed screw. Similarly, if there are three threads, there are three blades; and if four threads, four blades, and so on. Propellers generally have two or more blades, and this means that the screw is two or more threaded.

The *pitch* of the screw is the distance which the screw would advance in one complete revolution if it worked in a solid medium, such as steel or wood.

The *diameter* of the screw is the diameter of the circle described by the extreme point of the blades when the screw revolves in a fixed plane.

The *length* of the screw is measured along the axis, and is equal to the length of the blade measured parallel to the axis.

The *angle* of the screw is the angle contained between the blade and a plane perpendicular to the axis. It increases as the radius at which it is measured decreases.

The *disc area* is the area of a circle of which the diameter is equal to the diameter of the screw. More correctly, the disc area is less than the area, as defined above, by the area of the boss of the propeller at its largest diameter in a plane perpendicular to the axis.

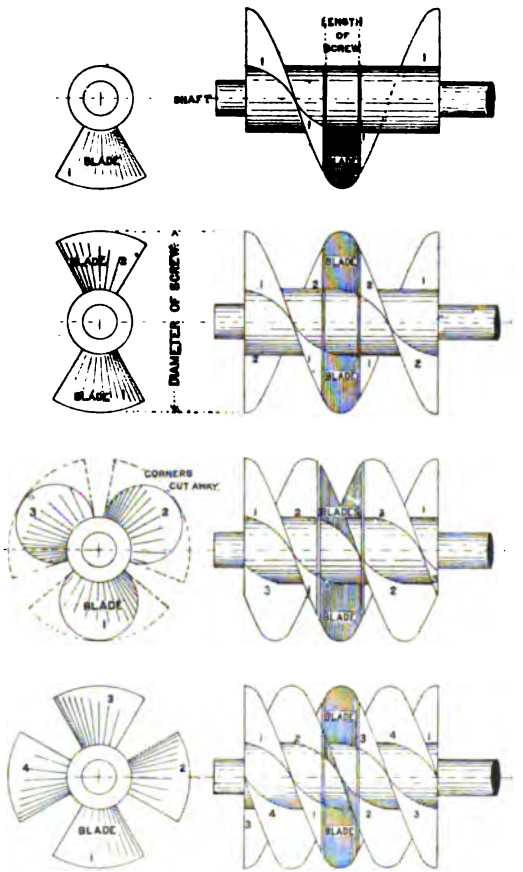


FIG. 286.—Screws and Blades.

The after side of the blade is called the *driving* face, while the forward side is called the *back* of the blade. The forward edge of the blade is called the *leading* edge, and the after edge is called the *following* edge.

The *developed area* of the blade is that measured on a plane parallel to the blade. The *projected* area is measured on a plane perpendicular to the axis, and is the developed area as projected on to this plane.

The *pitch-ratio* is the pitch divided by the diameter of the screw. When the pitch-ratio is great, the ratio of the projected area to the developed area is small.

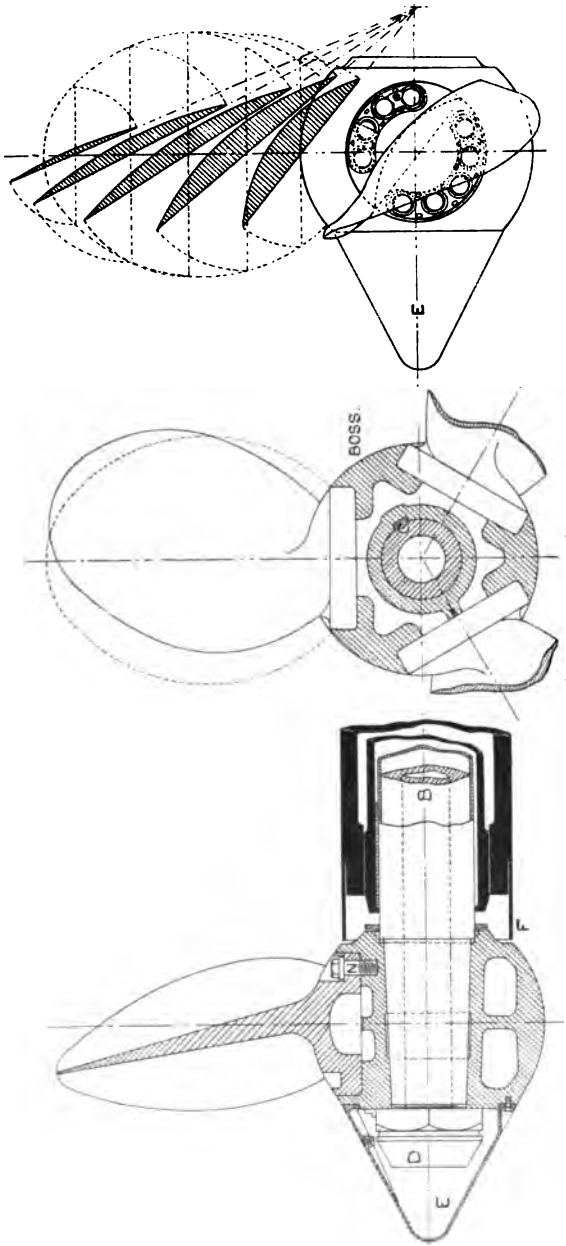


FIG. 287.—Screw Propeller.

Description of Screw. — Fig. 287 shows a longitudinal section through the boss and fittings of a large three-bladed screw; a section across the boss; and a plan of one blade with the bolt-holes and two bolts in place, and various cross sections of another blade, as used in some recent examples of naval vessels.

For turbine engines, the boss and blades are usually made in one piece of manganese bronze; but for reciprocating engines, the blades are generally made separately to the boss of the screw used for naval purposes and for mercantile vessels of large power. Manganese bronze is generally used for the blades, when made separately, because with this material they can be made thinner and thus offer less cutting resistance; but steel or cast iron is sometimes used when the blades and boss are made in one piece.

The boss of the screw is secured to the shaft *B* by a steel feather, *C*, and a cap-nut, *D*. The shaft is slightly coned, and the boss is driven or forced on and made a tight fit on the shaft. The cap-nut is prevented from slacking back by a guard plate, which is secured in the usual way by nuts and split pins on square-necked studs. At the forward end of the boss a packing gland is fitted, and generally consists of a flat ring, made in semicircular parts, with overlapping ends, pressed on to an india-rubber packing, slightly protruding from the recess into which it is jambed. The cap-nut and packing gland are fitted to prevent access of moisture to the shaft, between it and the boss, and from consequent corrosion.

A coned tail-piece, *E*, is fitted on the after end of the boss to cover the end of the shaft and nut, and to allow the water after passing between the propeller blades to take up a direct sternward and undisturbed flow without an eddying or cavitating wake. A guard, *F*, is fitted at the after end of the shaft bracket, and is shaped so as to allow a smooth run of water to the propeller; *F* protects the gland at the fore end of the boss, and also a guard ring, which keeps the *lignum vitæ* strips in place within the metal casing fitted inside the shaft bracket for carrying the shaft. The gun-metal casing fixed to the shaft to form its bearing surface extends to just abaft the india-rubber packing. The guard ring *F* is generally made in two parts, for convenience of removal, and sometimes of zinc, to protect other parts near it from corrosion. The gun-metal casing containing the strips is also made in two parts, and is generally designed to allow withdrawal from the fore end of the shaft bracket, without disturbing the propeller or shaft. As mentioned in Chapter XV., this casing must be removed before removing the shaft.

Variation of Pitch.—The blades of the propeller are flanged circularly near the root, and this flange fits into a space on the boss, accurately turned and fitted for the purpose. Gun-metal or bronze screws, *N*, secure each blade to the boss; the holes are elongated in the flange for the screws, and thus the blade can be set at a slightly different angle, with a corresponding difference of pitch of the screw propeller. This variation is convenient for slight alterations for trial, and also when the boiler pressure is necessarily reduced. This reduction of pressure is not common in modern practice; but, if decided on, the reduction of pitch allows an increase in the number of revolutions, and nearly the same power can be

developed, but with less economy, as with the original pitch and revolutions.

After the blade has been temporarily secured and tested for correctness of pitch, filling pieces are fitted in the recesses on each side of each securing screw; these are generally made of gun metal or *lignum vitæ*. Guard plates are fitted so that the screws cannot slack back, and there is generally one plate between each pair of boltheads. The plate is kept in place by the usual square-necked stud, through an elongated hole, nut, and split pin. A thin gun-metal covering plate is fitted over the boltheads and secured to the boss by countersunk screws. This plate is fashioned to present a fair surface to the passing water, and when closed up the boss of the propeller becomes almost globular in shape, except where the fore and after ends are flattened.

In the later and generally better shaped propellers great trouble is taken to secure—

1. A free and smooth run of water to the propeller and away from it, as it is important that the screw should act in water as little disturbed as possible.

2. A safe guarding of the securing arrangements of the propeller, its fittings, and surroundings. The proper inspection of these locking and securing arrangements is most important when a ship is docked.

For smaller screws with separate blades, the boss is made of steel, and the blades are secured by keying the root into a slot made diagonally across the boss. In this case the root of the blade should be made with a strong lip on the driving side, and the key should be secured by some positive method from moving and thus releasing the blade.

Slip of the Screw.—The screw propeller does not work in a solid medium, but in water which yields slightly, and has also been set in motion by the passage of the ship. The difference between the distance that the screw would advance if it worked in an unyielding medium and the distance that it is actually carried by the ship in the same time is called the *apparent slip* of the screw, and is that usually shown in tabulated results.

If p is the pitch of the screw in feet, and N is the number of revolutions of the engine *per minute*, then $p \times N$ is the distance the screw would have travelled in a solid medium in one minute; $(p \,.\, N)$ is called the speed of the screw.

If S is the speed of the ship in *feet per minute*, then—

The apparent slip $= p \cdot N - S$; or

The apparent slip $= \frac{p \cdot N - S}{p \cdot N}$, expressed as a *fraction* of the speed of the screw; or

$= \frac{p \cdot N - S}{p \cdot N} \times 100$, expressed as a *percentage* of the speed of the screw.

The *actual slip* is very difficult to obtain, because the screw is always acting in water already set in motion by its own action and the passage of the ship. It is evident that it must always be a *positive* quantity, or no thrust could be imparted; but in a few instances a *negative apparent slip* has been observed.

Negative Slip.—If the speed of the ship exceeds the speed of the propeller, the slip is said to be negative. In the equation quoted above, if the result is a negative quantity, the apparent slip is negative; and this is popularly known as negative slip, but is correctly negative *apparent* slip.

Apparent negative slip is obtained by the screw acting on the water set in motion by the ship, and following the direction of motion of the ship. With all propelled ships this following water, or following wake, is a usual occurrence. With ships with fine lines and suitable propellers, the following wake is not a serious loss, because the motion is imparted to a comparatively small column of water; but when the motion is imparted to a very large wake there is a great loss of propulsive effect, and although some of the energy imparted to the water is returned in assisting the propulsion of the ship, a much larger amount is lost. It is particularly when this loss is great that negative apparent slip is obtained, and generally it may be stated that this attainment must result in an actual loss of economy and efficiency.

From experimental data obtained from ships of very fine lines and high speeds, it is found that a constant, or slightly increasing apparent positive slip, varying from 10 to 20 per cent, generally gives good results; and, that the highest speeds are obtained with less power than with propellers which at the lower speeds showed an apparent positive slip of 5 to 10 per cent.

Direction of Rotation of Screws.—For vessels with a single screw the engine generally revolves with a right-hand motion when

going ahead, or in the same direction as the hands of a clock when looking towards it; the screw is then right-handed, and the tops of the blade move towards the right when looked at from aft. With outward turning *twin* screws the top ends of the blades move away from the ship's hull when going ahead, but with inward turning they move towards the hull. Inward turning, the port screw then being right-handed and the starboard left-handed, came into fashion for a few years (1895 to 1901); but outward turning has been reverted to. There is no proved advantage in efficiency of propulsion with either system; but inward turning allows the engine shafts to lie closer together, by placing the guide frames, of open fronted engines, on the outboard side of the shafting.

Comparative difficulties in manœuvring vessels are not due to the fact of screws turning either inwards or outwards, but rather to the relative position of the twin shafting. The shafts *diverge* from aft to forward with outward turning screws in many ships, whilst they *converge* with inward screws in others; (in H.M.S. *Drake* the lines of shafting produced cross each other almost exactly

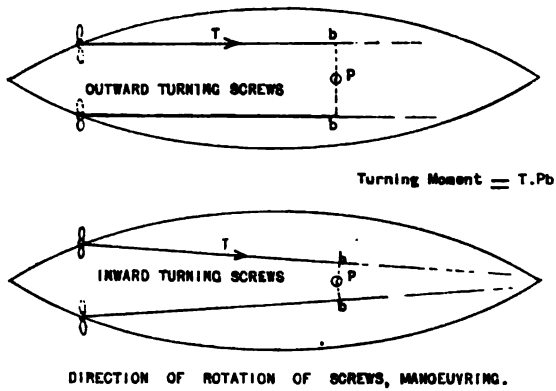


FIG. 288.—Manœuvring with Twin Screws.

at the stem). Vessels, when turning, pivot about some point, P (Fig. 288), in the keel line, which oscillates from somewhere near the bow to about two-fifths the length of the vessel abaft it. The difference in manœuvring is caused by a mechanical turning movement, $T \times Pb$, exerted along the respective shaft axes about the pivotal point. The outward turning screws being a little closer together and the engines farther apart the force exerted by the thrust T is conveyed to a long arm, Pb, while with inward turning the thrust is

conveyed to a short arm; this difference naturally results in better manœuvring power with parallel or divergent shafting. It is even possible with convergent shafting to pull the head of the ship to port when going astern with the starboard screw, the turning moment being then negative, with the pivotal point well forward. The above contention is in agreement with everyday practice, and there is reason to believe that the mere changing over of the propellers, port with starboard, would have no material effect on the handiness of any vessel; the difference lies entirely in the lines of shafting.

Twin Screws.—The diameter of the screw is limited by the draught of water and the greatest immersed transverse section of the ship. For ordinary sea-going purposes the tips of the blades should be immersed at least one-tenth the diameter of the screw, and sufficient clearance should be allowed both below and at the side to prevent the screw being carried away by grounding or going alongside another vessel or pier. If a single screw is employed, the flow of water to it is necessarily masked by a portion of the stern of the ship to which it is attached. The area acted on by the screw is very nearly in proportion to its diameter; and taking this into consideration, it is evident that a greater amount can be acted on with two screws than with one. Even if only the same area of water column is acted on by two screws as by one of larger diameter, there is still the effect of greater immersion and better propulsive effect of the two screws. A space of about 1 to $2\frac{1}{2}$ feet is generally allowed between the tips of the blades of twin screws and the walls of the vessel abreast the screws.

The advantages of twin screws may be considered as:—

1. Propellers can be of smaller diameter, the total disc area can be increased, and there is greater immersion with less racing; for warships there is less chance of disablement by shot and shell and floating wreckage.

2. Better run of water to the screws, with a possible increase of efficiency.

3. Manœuvring power.

4. Steering by screws alone: one screw at a constant speed, and the other varied as necessary to keep a course.

5. Duplication in case of breakdown of one engine.

6. Division of engine rooms by a water-tight bulkhead, adding to the strength and safety of the ship. This is a doubtful advantage unless the bulkhead is athwartships.

7. The engines can be smaller and of shorter stroke, and are thus under better protection.

Triple Screws.—When the beam is greater than three times the draught of water, the question of three or more screws must be considered if the power to be used and, therefore, the thrust to be imparted by the screws, cannot be obtained with twin screws. There appears to be no limit at present to the power which can be placed on one shaft by any available engine power within the limits of the vessel. The adoption of triple screws is therefore more dependent on the efficiency of the screw than on the possible efficiency of the engine.

For warships three screws may have a distinct disadvantage, because at low powers all the screws are unnecessary, and those not in use produce a drag which has to be overcome by those in use. The drag is decreased by about one-half if the screw or screws not in use are uncoupled from the engine and allowed to revolve freely; and a rough estimate of the power required to drag them round is perhaps about 10 to 15 per cent of the power developed in propelling the ship. Very little appears to be known on the subject of triple screws, and any opinion expressed must be accordingly of a general character

Quadruple Screws.—This arrangement has the comparative advantage of twin screws over a single screw, and has been generally adopted in preference to three screws in the largest turbine-propelled vessels (see Turbines).

Cavitation.—The propeller acts by giving the water a sternward velocity, the thrust being equal to the change of momentum, which is equal to the acceleration multiplied by the mass of water acted on per second. It is found that about half the velocity is given to the water before it reaches the screw, and about half after leaving it. The particles react on each other, and the force required to accelerate each particle is supplied by the difference of pressure before and abaft it.

Consider consecutive particles, P . . . Q . . . R, just forward of the propeller: then the pressure forward of Q is the same as the pressure abaft P, and therefore less than the pressure forward of P by the amount necessary to accelerate P; and the pressure abaft Q is less than the pressure forward of Q by the amount necessary to accelerate Q. Thus the pressure decreases as the propeller is approached, owing to the consecutive acceleration of each particle.

The pressure just forward of the propeller, in still water, is that due to the atmosphere *plus* the head of water above the propeller; and when the propeller is moving, the total head is diminished by that necessary to accelerate the particles of water forward of the propeller—that is, the reduction of pressure just forward of the propeller cannot be greater than the sum of that due to the atmosphere and to the head of water above the propeller. Therefore, if half the thrust pressure per square inch on the blades (*i.e.* the equivalent of a reduction of pressure on the forward side of the blades) is equal to that due to the total head, the pressure just forward of the blades becomes *zero*. At *zero* pressure the water will "break," and cavities are formed forward of the blades. This is called *cavitation*.

Cavitation, principally due to centrifugal action, can also occur abaft the screw, and, to counteract this, in a few cases where a very high speed of rotation of the propeller is used, the conical tail-piece abaft the screw is fitted with three (or more if required) vanes, which are nearly parallel with the axis of the screw. The vanes are fitted a little abaft the greatest diameter of the tail-piece, with their outer edges nearly parallel to the axis. The reason for fitting these vanes is based on the following assumption: The angular velocity of the water rotated by the screw blades is at least equal to that of the blades. The column of water to which this rotation is imparted decreases in diameter as it leaves the screw, closing in with the gradual decrease in diameter of the cone; but the velocity remains about the same, while the angular velocity increases, and thus becomes greater than that of the vanes, which revolve with the propeller; therefore the water presses on the vanes and tends to increase the rotation of the shaft, while the reaction tends to reduce its own rotative motion, and, increasing the pressure on the tail-piece, lessens the tendency to cavitation.

To avoid cavitation, the following conditions must be fulfilled:—

1. The projected surface of the screw blades must be sufficient.
2. The screw must be sufficiently submerged.
3. There must be fine lines to give a clean run of water to the propeller; because a greater suction is required to draw the water to the propeller if the run be not sufficiently fine.

Thrust Pressure.—For any form of screw, and shape or form of vessel, there is some definite limit to the speed at which water reaches the propeller, and this limit may be considered to define the

thrust pressure which can be imparted by the screw blades without causing cavitation. Cavitation, then, will take place, as we have seen, when the *projected area* of the blades is so small that the suction thrust—which may be taken as one-half the total thrust—per unit of area is greater than the atmospheric pressure *plus* the head of water above the screw.

The head of water may be usually neglected; thus the limiting thrust per square inch on the projected area may theoretically be considered as twice the atmospheric pressure, or 30 lb. per square inch. If the screw be sufficiently submerged to make the head appreciable, an addition of $\frac{3}{8}$-lb. per square inch can be made to the thrust pressure for each foot-depth of immersion of the tips of the blades. In practice, a thrust pressure of 30 lb. per square inch cannot be realised; for, of course, cavitation takes place before zero pressure is reached, on account of the air and vapour which is held in suspension or solution in the water becoming freed at a low pressure.

The practical limit of thrust pressure with screws of ordinary form and shape, such as have been found to be efficient for fast-running engines and high speeds, is about 10 lb. per square inch on blades submerged about one foot. This thrust is that on the *projected* area of the blades; the *developed area* depends on this projected area and the pitch-ratio adopted.

The trials of H.M.S. *Drake* furnish a good example of the beneficial advantage of increasing the blade area when the thrust is excessive.

TRIALS OF H.M.S. *DRAKE* (1902)

	1st Screws.	2nd Screws.
Diameter	19 feet	19 feet
Pitch	24·5 feet	23 feet
Pitch-ratio	1·29	1·21
Expanded surface of the blades, each screw	76 sq. ft.	105 sq. ft.
Projected surface of the blades, each screw	58·5 sq. ft.	82·7 sq. ft.
Revolutions per minute	116	122·4
Speed in knots	23·05	24·11
Apparent slip per cent	18	13·35
Thrust pressure per sq. inch of *projected* area, assuming effective thrust = 60 per cent of indicated thrust	15·1 lb.	10·2 lb.

Surface Friction of the Blades.—The frictional resistance of

the blades is of great importance; it depends on the condition of the surfaces, their area, and the velocities with which the various units of their area cut through the water. It is evident that a square foot of blade area near the tip will cut the water at a much greater velocity than a square foot near the boss, and that the frictional resistance in consequence will also be greater. The form of blade usually adopted until recently was elliptical, with the minor axis about four-ninths of the major axis, which extended from the tip of the blade to the centre of the shaft. If the same blade area is used and the minor axis is increased to about six-ninths the major axis, the diameter of the propeller can be decreased. This alteration may produce a decreased frictional resistance, and at the same time an increased efficiency from an increased immersion of the tips. Some results tend to prove the practical efficiency of this alteration; but it must not be forgotten that any diminution of diameter also diminishes the area of the column of water projected astern by the propeller, so that now the screw is generally made of the largest possible diameter which can be efficiently immersed and suited to the engine speed, while the area is disposed as near as possible to the boss, consistent with a fair curvature of the edges of the blade, as shown in Fig. 287.

In warships particularly, it is important that the blade area should not be unnecessarily increased, because the extra surface friction reduces the economy at low powers. In all cases the blades should be cleaned at every opportunity, and polished on the surface, to reduce the frictional resistance; and an increase of over a knot in speed has been frequently obtained by proper attention to this detail when in dock. With the great speed of rotation now used for warships and turbine engines, the condition of the surfaces of the blades becomes of increased importance.

Projected Area of Propeller Blades.—For the speed of vessel required, the I.H.P. or S.H.P. and the number N of revolutions per minute and the pitch of the propellers can be calculated from previous practice. The effective H.P. may be taken as from 45 to 55 per cent of the I.H.P. (call this percentage K), and the thrust pressure on each square inch of projected area of the propeller blades is limited by practical considerations of cavitation, etc., to 10 lb.; then—

$$\text{Total effective thrust} = \frac{\text{I.H.P.} \times 33{,}000 \times \text{K}}{\text{pitch} \times \text{N}} \text{ lb.}$$

$$\text{Projected area in sq. ft.} = \frac{\text{I.H.P.} \times 33{,}000 \times \text{K}}{\text{pitch} \times \text{N} \times 10 \times 144}$$

$$= \text{K} \times 23 \times \frac{\text{I.H.P.}}{\text{pitch} \times \text{N}} \text{ nearly.}$$

$$\left.\begin{matrix}\text{For turbine engines} \\ \text{(and S.H.P.)}\end{matrix}\right\} \text{K} = \text{about } 65 \text{ per cent} = 0{\cdot}65.$$

Turbine-screw Propeller.—A type of Thornycroft turbine propeller is shown in Fig. 289, and is sometimes used for shallow-draught vessels.

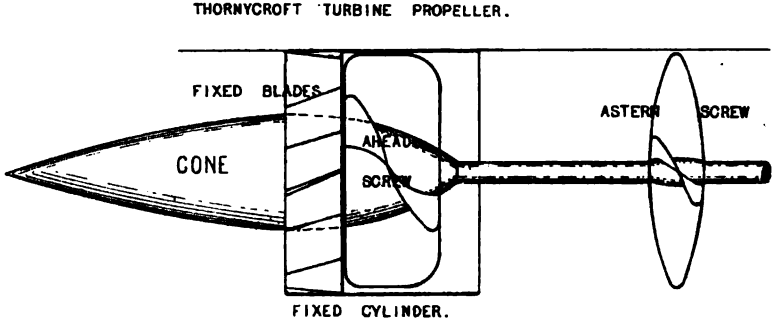


FIG. 289.—Thornycroft Turbine-Screw.

The ahead screw is of small diameter, and revolves inside a cylinder fixed to the vessel. The revolution of the ahead screw, which is of increasing pitch, produces some rotation in the column of water passing through it. This rotation is reversed by fixed guide blades fitted inside the cylinder just abaft the screw, and the column of water is projected directly sternward without rotation. There is some loss of efficiency from the friction on the guide blades, but it appears to be small compared with the extra thrust obtained per unit of area of the blades. At the after end of the guide blades there is an elongated cone, of the same type as fitted to the ordinary screw propeller; but it is generally longer, and the water thus takes up its sternward motion with the least possible disturbance and no eddying wake.

The pitch at the fore end of the ahead propelling blades corresponds approximately to the velocity of the water entering the screw, and the velocity of the water passing through is accelerated by the gradually increasing pitch of the blades towards the after

end. As the flow of water is accelerated during its passage the area through which it passes is gradually decreased by the front part of the cone, through both the propelling and the guide blades.

This type of propeller cannot be efficiently used for going astern, because the column of water entering the guide blades from the after end cannot be sufficiently accelerated, and therefore little useful thrust can be imparted. For going astern another screw is fitted on the same shaft, as shown, at some little distance in front of the ahead screw, and of the same diameter and pitch as the forward end and leading edge of the "ahead" screw. When going "ahead" the astern screw imparts no thrust, and simply revolves with the shaft.

Mr. Barnaby estimated the thrust produced by the guide blades at about one-third of the total thrust of the propeller, but that the extra friction absorbed sufficient power to reduce the efficiency to a little less than that of the best ordinary screw. A thrust pressure of about 26 lb. can be allowed on each square inch of projected area of the turbine screw, with about the same efficiency as the ordinary screw propeller.

The turbine screw is not necessarily totally immersed when the ship is at rest. It can be fitted in a tunnel of suitable shape, so that after a few revolutions of the screw all the air is expelled, and the screw then acts in solid water.

Yarrow Shallow-draught Arrangement.—When a boat designed for shallow draught is loaded down considerably, the water passing through the propeller cannot easily escape, and there is con-

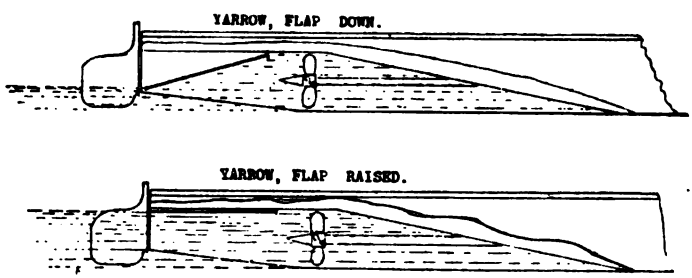


FIG. 290.—Yarrow Shallow-draught Tunnel.

sequently a drag on the vessel, set up by the pressure exerted by the projected water on the after end of the tunnel. To overcome this difficulty, Mr. Yarrow has introduced an arrangement by which the

after end of the tunnel can be raised or lowered. A horizontally hinged flap, partly forming the roof of the tunnel, is fitted just abaft the screw, and its trailing edge is raised or lowered as necessary to keep it at, or just below, the surface of the outgoing water. In Fig. 290 the flap is shown down in one view, as used in normal state of draught of water; and in the other the flap is raised for abnormal draught. A distinct and large gain in speed was obtained with this arrangement, when the boat on which the trials were made was loaded down considerably. The flap must be sufficiently raised when going astern, to allow a free flow to the propeller.

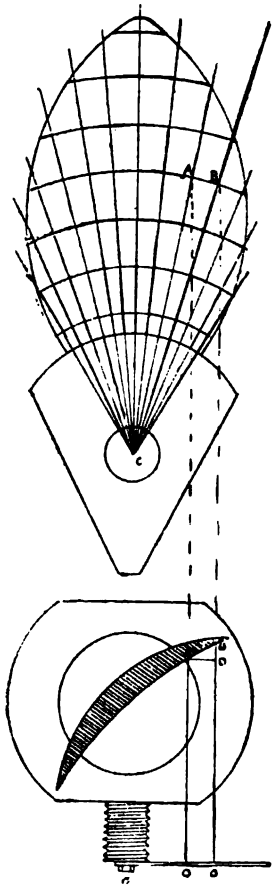


FIG. 291.—Measurement of Pitch of Screw.

Measurement of the Pitch of the Screw.—The pitch of the screw propeller is measured by placing the completed screw, set at about the required pitch temporarily, on a large face-plate, with one of the faces of the boss resting on the plate, or parallel to it. The driving face is preferably placed towards the plate, and then all measurements can be taken directly from it. In this manner a more perfect reading of the various pitches can be obtained than in the method generally used after the screw is fitted in place on the shaft, and after its measurement in this accurate way any small deviations can be removed by grinding the face to a truer surface.

When the screw is on the shaft in dock the method of measurement is as follows (Fig. 291):—The tail-piece at the end of the shaft is removed, and a radius bar, *CO*, fixed at the end *C*, fitted to the end of the shaft. The point *C* is made coincident with the centre or axis of the shaft, and when the bar is moved it should always lie in a plane perpendicular to the shaft. A number of parts of circles, at different radii, pitched not more than one foot apart for large propellers, are described on the propeller blade by means of a bar, *OA*, with

a sliding connection, which connection can be clamped at any required radius on *CO*. The bar *OA* must always be parallel to the axis of the shaft, so that the circle described from *C* as centre has *CO*, equal to *CA* in the upper figure, as radius. Denote any radius, *CA*, by *r*. By means of a circle on the end of the shaft or propeller boss, set off radii at equal angles. Call each of these equal angles, through which the radius bar *CO* is moved, θ.

The pitch is found for every point where the radii cut the circles; and the *mean* of the pitches thus obtained is called the *mean pitch.*

EXAMPLE.—Suppose the pitch is required at any point, A.

Then measure OA and OB accurately, from which referring to the lower figure, BD = OB – OA. In one complete revolution the point O describes a circle, the circumference of which circle is $2\pi r$, of which AD is a part. This circle is cut into $\frac{2\pi r}{AD}$ equal parts by the radii. Also in one revolution the point A travels, in a direction measured parallel to the axis, a distance equal to the pitch of the screw, which may be divided into $\frac{2\pi r}{AD}$ equal parts, each part being equal to BD.

Therefore the pitch at the point A (or practically at any part AB, if A and B are near together)—

$$= BD \times \frac{2\pi r}{AD}, \text{ where } \pi = 3{\cdot}1416.$$

$$\text{Or, the pitch} = \frac{BD \times 2\pi}{\theta}, \text{ where } \pi = 180^\circ, \text{ and the angle } \theta \text{ is known.}$$

For example, let $\theta = 7\frac{1}{2}^\circ$; then the pitch—

$$= BD \times \frac{360}{7\frac{1}{2}}$$

$$= BD \times 48.$$

That is, the pitch is 12 inches for every $\frac{1}{4}$-inch of BD.

A *more approximate mean* is obtained by calculating the mean pitch at each radius and taking the mean of the means thus found. The greater the number of radii employed, the nearer will the approximation be to the correct mean. Generally, this second method of calculation after measurement will give a smaller mean pitch than that obtained from the previous method.

Chapman-Hunter Pitchometer.—This small instrument, which can be carried in the pocket, can be used for measuring the pitch of any sized propeller without calculation, either in the shop or in position on the shaft. Lines are drawn at various radii on the blade from the centre of the boss, say at intervals of 1 foot. The

instrument is shown in Fig. 292, and consists essentially of a plate, *P* (with feet, *KK*), on which scales are engraved for various

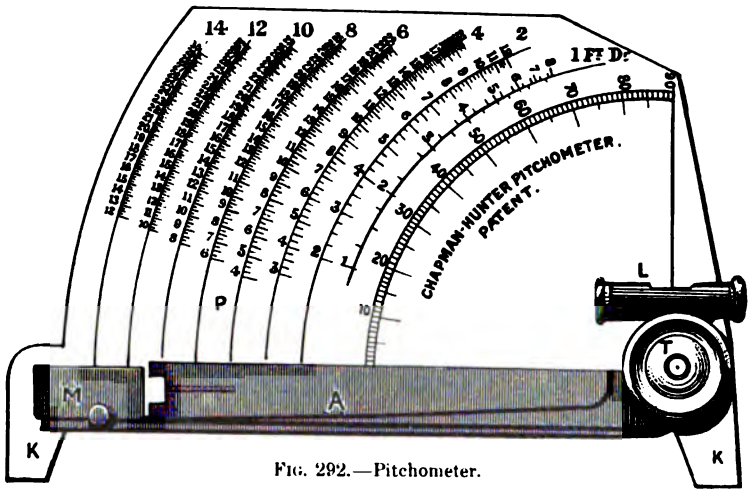


Fig. 292.—Pitchometer.

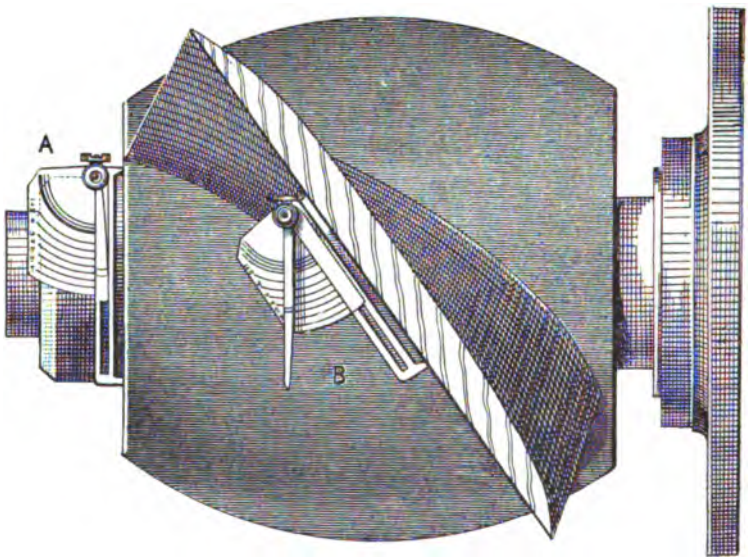


Fig. 293.—Application of Pitchometer.

diameters; an adjustable spirit level, *L*; an adjustable arm, *M*; and a thumb screw, *T*, for adjusting the position of the level and at the same time securing it to the arm.

If the propeller be in the position shown in Fig. 293, the instrument is first placed in the position *A* with its feet resting against the face of the boss ; the arm is kept closely pressed into its zero position, and at the same time the level is set until the bubble shows exactly at the middle line ; and the level and arm are then secured by the thumb screw, so that they cannot alter their relative positions. Next, the instrument is placed at *B* with its feet resting on a pitch line ; the arm is adjusted until the bubble again shows in the mid-position, and the pitch, corresponding to the cross lines at the edge of the arm and diameter of the pitch circle, can be read on the scale. For blades whose tips slope sternwards, and overhang the boss and tail-piece, it is convenient to turn the shaft as necessary to measure the downcoming blade, starting with the smallest diameter of pitch circle, and turning the shaft as necessary for each pitch circle until all are measured. In this way the pitchometer remains plumb and gives correct results.

Disposal of the Pitch.—There are three methods in general use of disposing the pitch :—

1. The most general is to use an absolutely true thread, with the pitch uniform throughout. This is commonly used in the Navy and the merchant service, but there are a few exceptions.

2. The pitch is made somewhat larger towards the following edge, and also increasing towards the tips of the blades. It is thought that a greater acceleration can be given to the water as it passes through the screw by this increasing pitch, and it appears a reasonable theory.

3. The pitch is decreased towards the tips of the blades with a view to reducing cavitation, which generally commences at the tips of the blades.

At present there is insufficient evidence on any one of these to confirm its superiority to any other. In some large German liners the driving faces of the blades are machined to an absolutely true designed pitch, which is generally of the slightly increasing type. There is some advantage in this practice, but whether the advantage is due to increased accuracy or to reduced friction is a matter of some doubt. On the other hand, Messrs. Harland & Wolff, and probably other firms, trust to obtaining blades of exactly similar form by first making a cast-iron blade from the wooden pattern, and then using the cast-iron one as a pattern for the actual blades made of manganese bronze. The surfaces of the blades are ground fair, and

the leading and following edges are carefully chipped or filed to clean cutting edges. The surfaces are seldom painted, and the propellers give apparently good results. With fast-running screws driven by turbines or high-speed naval engines, a polished surface increases the efficiency, and incidentally it may be mentioned that this practice was initiated in the Navy by the author about 1895 in H.M.S. *Decoy*; non-painting is now embodied in the Steam Manual (1906).

PART VIII

AUXILIARY MACHINERY

CHAPTER XXVI

THE STEERING AND CAPSTAN ENGINE AND GEAR

Steering Mechanism.—Some mechanism for controlling the helm is an absolute necessity for large or fast vessels. The pressure on the rudder is very great when the ship is turned, and, without mechanism, is beyond control by manual labour. The time required to move the rudder is proportional to the power which can be employed in moving it, and consequently the first consideration, in designing a mechanism for the purpose, is to meet the time requirements from midship to extreme helm (about 10 seconds), and from extreme helm to extreme helm (about 20 seconds). The mechanism is generally constructed in three principal parts:—

1. The controlling gear, worked by the helmsman at any required point in the ship, which works a valve of the differential reversing type.

2. The engine, which obeys the movement given by the helmsman, through the controlling gear, to the differential reversing valve.

3. The mechanism or gearing, which conveys the power of the engine to the crosshead or tiller, rigidly connected with the rudder.

Controlling Gear.—A wheel is generally used for moving the controlling shafting, but in a few cases a small lever is used. The lever, or upper part of the wheel in naval steering gears, is arranged to move in the same direction as the bow of the ship

when it responds to the movement of the rudder. The wheel mounted on a horizontal axis, *A*, within a vertical pedestal, *B* (shown in Fig. 294), rotates a vertical shaft, *C*, through bevel gearing wheels, *DD*. The shafting is continued by horizontal and other necessary lengths, connected by bevel wheels, or Hooke's joints, with a lever in connection with the differential valve spindle.

The number of turns of the wheel is limited, generally four revolutions from amidships to extreme helm. Inside the pedestal the shaft is screwed with a properly pitched thread, *E*. A certain number of turns of the steering wheel revolves the screwed shaft through a nut, *F*, which is moved along, but does not revolve with the shaft. A projection is made on the bevel wheel at *K* and on the collar *G*; and the wheel *DK* and collar *G* being keyed to the horizontal shaft, revolve with it. At the extreme points of the helm allowed, projections on the nut *F* come in contact with the projections on *DK* and *G*, and further rotation is prevented. A cogged rack, *R*, made in one piece with the nut *F*, moves with the nut and rotates a cogged wheel, to which a pointer, *P*, is attached; the pointer *P* is outside and generally on top of the pedestal. Two dials, one engraved on the top and the other printed on glass (so that a light placed inside the pedestal shows through it), show the amount of helm, in turns of the wheel and in degrees, imparted by the controlling gear. This index does not necessarily show the exact position of the rudder or tiller in connection with it; this is shown by the *helm indicator*, which is mechanically geared to the rudder head. If the controlling gear, engine, and rudder are properly connected to work in unison, then the pointer on the steering pedestal indicates the position of the *rudder*, as well as the controlling gear.

The details in connection with the controlling shafting are of great importance, as the safety of the ship frequently depends on its efficiency. Generally the shafting is hollow and stiffened at the couplings by screwed plugs, fixed by rivets, as shown in the various figures. The weight of the shafting, if horizontal and of any length, is carried on roller bearings; the bevel wheels are carefully machined to gear into each other with as little back lash as possible, and without undue friction; and the keys and pins securing the wheels to the shafting are fitted with stops or split pins to prevent them working out of place.

The various lengths of shafting are connected together by sleeve

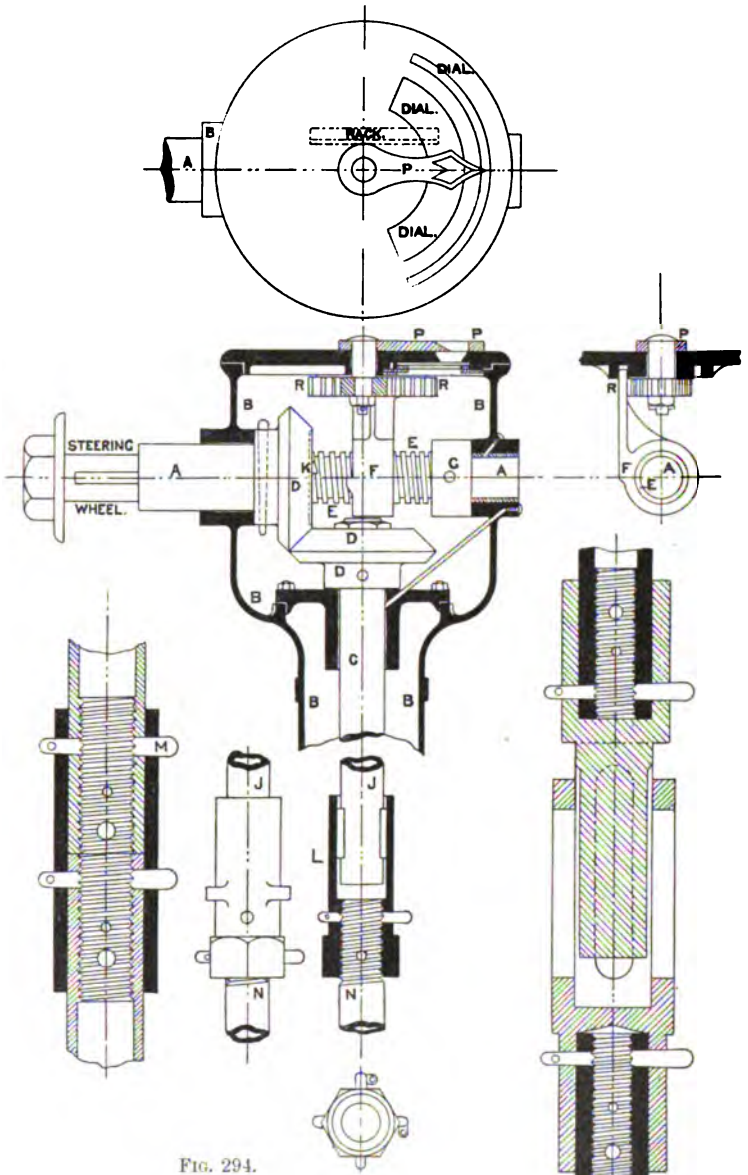


FIG. 294.

FIG. 295.

FIGS. 294 and 295.—Steering Control Pedestal and Shaft Couplings.

pieces, secured either by taper pins, *M*, or by cotters, or by small bolts and nuts, with split pins to prevent them falling out. The cotters at various parts of the shafting are free to move through elongated holes in the sleeve piece, and thus allow for expansion. In some cases squared ends, working in squared sockets, are used for expansion couplings; but some stop should then be fitted to prevent the two parts of the shaft disconnecting under a sudden stress or alteration of form of the vessel. In a destroyer, such disconnection has taken place in very rough weather. Unless the deviating angle of the shafting is very small, bevel gearing is used in preference to a Hooke's joint. When steam is on the engine, a movement of less than two spokes of the steering control wheel, or one-eighth of a turn, should set the engine in motion.

If the steering wheel is run over very quickly, the engine following in unison with it, the momentum of the engine and the heavy parts moved by it are liable to carry the tiller over farther than the stops inside the pedestal would otherwise allow. The stops inside the pedestal are therefore generally set two or three degrees within the extreme range of the tiller. A connection is fitted to the rudder head or tiller, as convenient, which, coming against projections attached to the ship, acts as a stop in each of the extreme positions—generally about 35° each way from the hard-a-port or hard-a-starboard position to the middle position.

In warships, pedestals and steering wheels are fitted in various parts of the ship, and each is generally connected by shafting and a coupling with one of the main runs of shafting from the foremost steering positions. As far as possible the controlling shafting is kept below the water-line, under protection from shot and shell, and the couplings for the various steering positions are therefore generally in this position. The end, *J*, of the controlling shaft leading from any steering position, *C*, is generally connected with the main run of shafting by screwing into a sleeve permanently attached by a sliding joint, *L*, to the main run by a short length of shafting, *N*, and bevel wheels. When the two parts of the shaft are in coincident positions with the same helm, as shown by their respective dials at the steering positions, the position of the end of the shaft is such with respect to the sleeve that it can be connected with it by inserting two taper pins, prevented from working out by split pins. This arrangement is necessary to make it impossible to connect up the various steering positions in incorrect positions

relatively to each other, and they can thus be connected correctly when communication with the various positions is impracticable.

Brotherhood's Coupling.—The object of this coupling is to enable two shafts to be coupled or uncoupled at any time when one or both are being rotated; the same end is attained in the screw coupling previously described, but the Brotherhood arrangement is handier and more convenient. Referring to Fig. 296, *A* and *B*

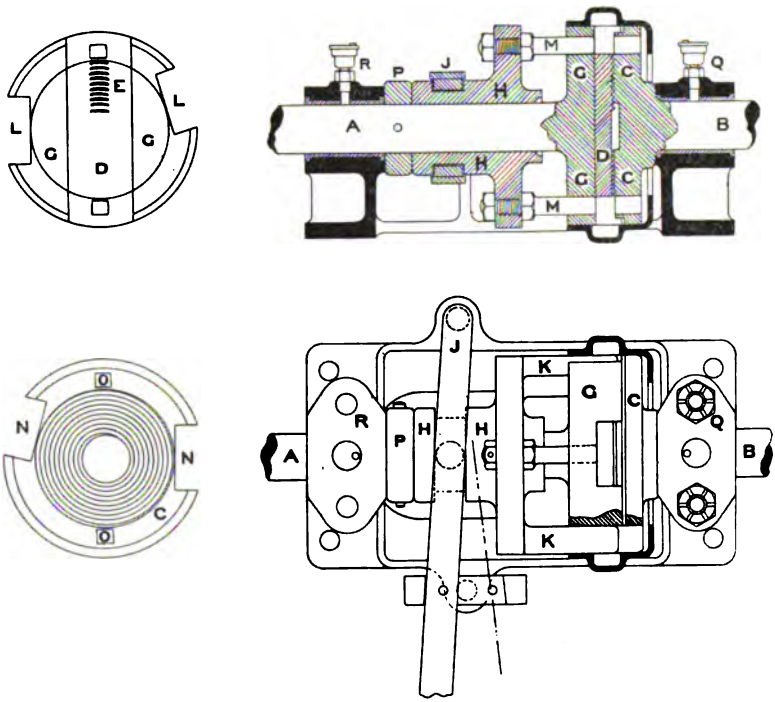


FIG. 296.—Brotherhood Coupling.

are the two shafts to be coupled. A flange, *G*, is made on *A*, and in a slot made in the face of *G* there is a sliding piece, *D*, with projecting teeth, *E*, which is constrained to move in the slot. The teeth *E* engage in a spiral projecting thread formed in the face of the opposite flange *C* of the shaft *B*. The figures show the uncoupled position; and when either *A* or *B* is rotated independently of the other, the sliding piece *D* moves to and fro in the slot corresponding to the motion imparted by the projecting teeth *E*. *H* is the coupling, keyed to the shaft *A*, and capable of being moved

along *A* by the hand lever *J*. On *H* there are two projecting lugs, *K*, always engaged in the spaces *L* formed in the rim of the flange *G*, and which when the lever is pushed to the right also engage in the spaces *N* of the flange *C*, and thus form the coupling between *C* and *G*. The sliding piece *D* prevents the coupling being moved to the right, except when in the correct coupling position, by preventing the pins *M* fixed in the coupling from being moved to the right. When the shafts are in correct relative position the pins *M* enter holes, *O*, near each end of the sliding piece *D*, and almost simultaneously the lugs *K* engage in the opposite spaces *L*, thus completing the connection. The spaces *L* are not diametrically opposite each other, so that the connection cannot be made half a turn out of the correct position. The teeth *E* and spiral thread are always engaged, and longitudinal movement of the two shafts, which would throw them out of gear, is prevented by a shoulder on *C* and a collar, *P*, fixed to the shaft *A*, bearing against the ends of the shaft bearings *Q* and *R* respectively.

Brown's Hydraulic Telemotor.—In some cases small pipes conveying hydraulic pressure have been substituted for the mechanically geared controlling shafting. The arrangement consists, as shown in Fig. 297, of a hand-steering wheel which, through a series of cogged wheels, *G*, *H*, *I*, *J*, *K*, and shaft, *E*, and pinion, *D*, moves a cogged rack, *C*, attached to a piston, *B*, working in a cylinder, *N*, at the upper part of the pedestal *O*. The piston is packed with double leathers, and when in the middle of its stroke, free communication is allowed, through an enlargement, *M*, of the cylinder, between the upper and lower sides of the piston. The cylinder is kept fully charged with water, mixed with about 30 per cent of glycerine, to prevent it freezing. Pipes lead away from the top and bottom of the cylinder to another cylinder, *R*, called the *motor* cylinder, fitted near the steering engine. The piston *S* in the motor cylinder is connected with the differential reversing valve. Any movement of the wheel tends to compress the water in one end of the first cylinder, and, forcing it through one of the connecting pipes, moves the piston in the second cylinder, and thus moves the differential valve and sets the engine in motion. The water expelled from the second cylinder is returned to the first cylinder through the other connecting pipe. The resulting motion of the engine, when the differential valve is moved, returns the latter to its central position and stops the engine, as in other types.

When the steering wheel is put amidships, free communication is allowed between the top and bottom of the piston in the first cylinder, as already explained, and the piston in the second cylinder

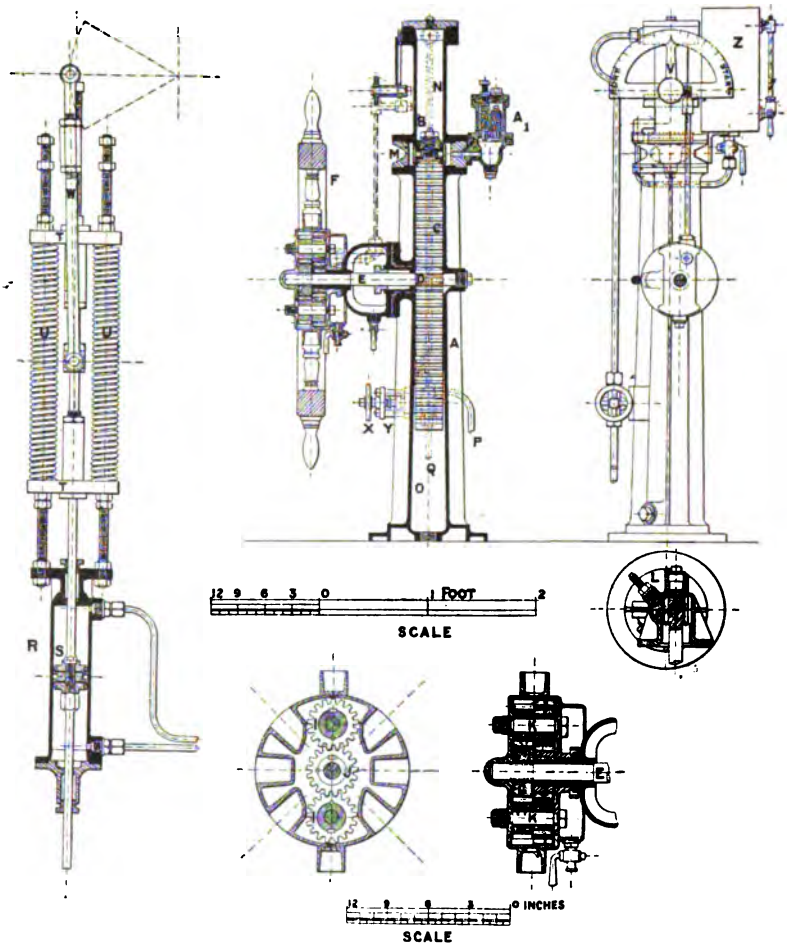


FIG. 297.—Hydraulic Telemotor Controlling Gear.

is returned to the middle of its stroke by two springs, *U*, either one of which is compressed when the piston is moved away from the middle of its stroke. Consequently, whenever the steering wheel passes the amidships' position, any inaccuracy between the relative positions of the two pistons tends to be adjusted. The first cylinder

is twice the capacity of the second, and there is always sufficient head to overcome the frictional resistance of the passages.

Differential Reversing Valve.—By reference to Figs. 167 and 168, it will be noticed that when the crank and eccentric arms are in the positions *OP* and *OS* respectively, the direction of rotation

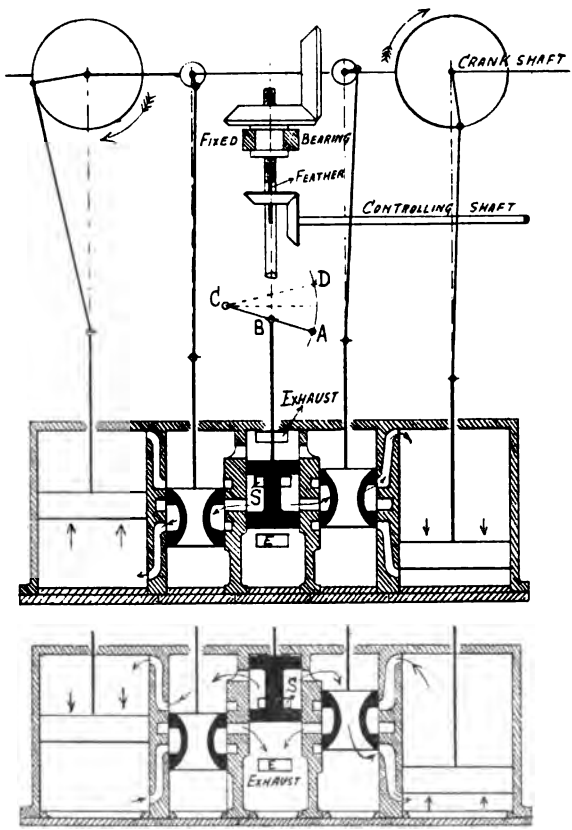


FIG. 298.—Hunting Gear.

of the shaft is the same as that of the hands of a clock. Now, consider the direction of motion when the steam is admitted through the exhaust cavity, and that exhaust takes place at the outer edges of the valve; then the direction of rotation is reversed, and becomes anti-clockwise. The valve used to obtain the changing over of the ports of entry and exhaust of the steam is called a differential reversing valve, and is the middle one of the three valves shown in

Fig. 298, which are shown as cylindrical valves with an inner hollow space connecting the spaces at the ends of each respective valve.

The engine is fitted with a pair of cylinders (see also Fig. 299), and the two cranks are set at right angles on the shaft, so that there is continuous motion and the engine can be started from any position of rest. Steam enters the differential valve chest through a port, *S*, and the exhaust steam leaves by the ports *E*, which are connected with each other and the exhaust pipe. The movement of the steering control shaft operates on a pair of bevel wheels, one of which rotates the differential valve rod by means of a feather. This rod takes the place of the lever *ABC*; and its rotation either screws it further into or out of one of a second pair of bevel wheels, which rotate in a fixed bearing and with the crank-shaft respectively. Suppose the movement of the control shaft has placed the differential valve in the position shown in the upper figure; then the engine is set in motion and revolves the crank in the direction shown by the arrows, and, rotating the upper pair of bevel wheels, the rod and valve are brought to the central position by the rotation of the crank, and the engine is brought to rest by the automatic cutting off of the steam-supply. Similarly, if the control shaft be rotated in the opposite direction, the crank-shaft is also driven in an opposite direction and automatically cuts off the steam-supply and stops the engine.

The gearing which effects this automatic stopping is known as *Hunting Gear*, and is commonly fitted for steering engines in connection with the differential reversing valve; this valve is used for many engines in which economy is of little importance compared with a quick and certain movement and reversal. The principal ones are the steering and capstan engines, hoisting engines generally, whether driven by steam or water pressure, and the turning and reversing engines used in connection with the main propelling engines; these engines are only occasionally, not continuously, in use for short periods, and economy is sacrificed to obtain practical utility.

Steering Engine.—In Fig. 300 three views in part section are shown of one of a pair of steering engines fitted in recent battleships; each engine is an exact duplicate of the other, and by suitable couplings either one can be used as required. Each engine has two cylinders, *H*, with cranks fitted at right angles to each other on the shaft *A*. The differential reversing valve *F*, and slide valves

G, are of the solid type, with open communication between the respective opposite ends of each valve through passages in the casting. Steam enters through the pipe *S* and surrounds the middle part of the valve *F*. Exhaust steam leaves through the passage and pipe *E* and fills the spaces above and below the valve *F*, which is moved up or down by the rotation of either the control shaft *C* or the hand wheel *N*; *C* and *N* are usually connected, and

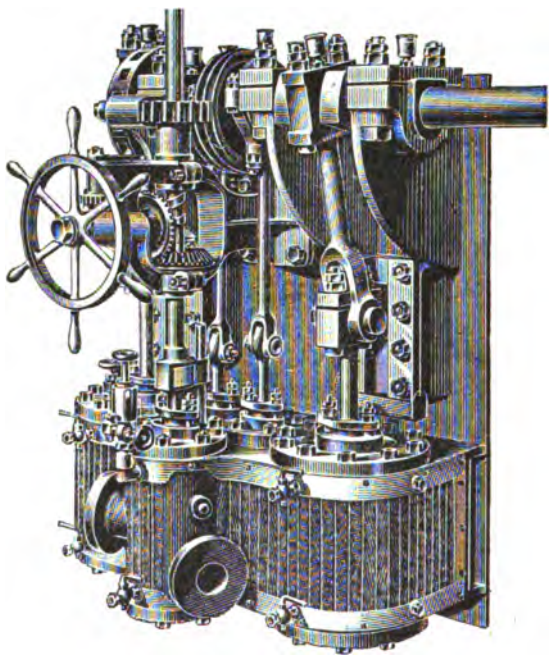


FIG. 299.—Steering Engine.

rotate together. Rotation of the shaft *CN* screws it through the centre of the worm wheel *D*, which revolves in a fixed bearing, *Q*, and raises or lowers the valve rod *B* and thus admits steam to the slide-valve chests and sets the engine in motion. By means of a worm on the shaft *A* gearing in the wheel *D*, the rotation of the shaft automatically shuts off steam, through the valve *F*, and stops the engine.

The framing *P* of the engine is usually secured to the engine room, or other, bulkhead. At *M* there is a hand-turning wheel when steam is not on the engine. *K* is an index (showing

the relative position of the steering control wheel), which is moved by the spur gearing *L* and rotation of the control shaft.

A slight lap is necessary for the differential and slide valves to prevent leakage, and consequently the steering control shaft and wheel must be moved a little before a cylinder is supplied with steam and the engine begins to move; generally one-eighth of a turn overcomes the back lash in the gearing and the effect of lap.

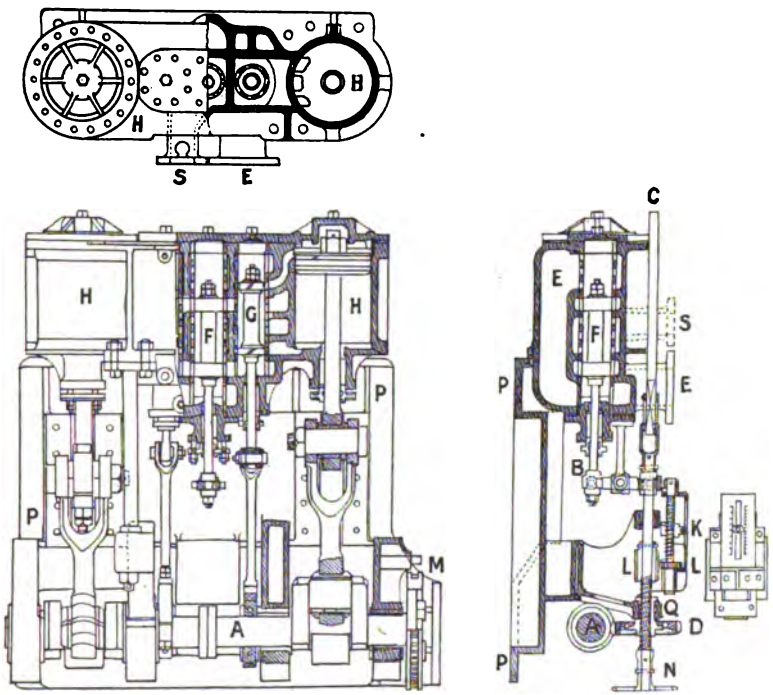


FIG. 300.—Steering Engine and Differential Valve Gear.

Rapson's Slide.—In this arrangement the tiller is worked by wire ropes, or chains, or by pitch chains running over spiked wheels. The rope or chain is worked by the engine, and the two ends are connected at opposite sides of the end of the tiller with a carriage or slide. The chain forms an isosceles triangle, with an angle below a horizontal base; the chain is led over a pulley at each angle next the base, and, running horizontally, connects with the carriage and completes the base line of the triangle. The carriage either moves along a sunk slide or moves along on rollers attached

to itself. The end of the tiller projects through a swivel block in the carriage, and as the tiller is moved over, the end slides to and fro through the block.

This gearing has the mechanical disadvantage of not increasing the force applied to the rudder in proportion to the greater degree of helm imparted by the engine; an ideal arrangement should have an increasing effect. Wire ropes are still used for steering, and chains are common in small vessels, but are generally less reliable than the arrangements next described.

Napier's Differential Screw Gearing.—A plan of this gearing, which is fitted in a great number of vessels, and has been in common use for over forty years, is shown in Fig. 301. The movement of the engine rotates the shaft *A*, on which square-threaded

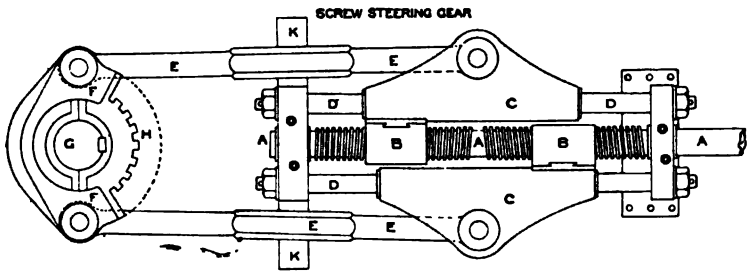


Fig. 301.—Napier's Differential Screw Gearing.

right- and left-handed screws are cut. The screws work in gun-metal nuts, *B*, attached to arms, *C*, sliding along fixed bars, *D*, and connected by rods, *E*, with a crosshead, *F*, attached to the rudder head *G*. The rotation of the shaft moves the nuts either closer together or farther apart, and thus moves the rudder. This gearing is self-holding, and no stops or pawls are necessary.

Sufficient end-play must be allowed for the screwed shaft in its bearings, and in its connection with the engine shaft, for the variation caused by obliquity. Thus when the arms are brought together, putting the rudder to port, the rod in connection with the left arm should be shorter than the other rod, and *vice versa*.

Harfield Steering Gear.—The engine is of the usual type, and drives a shaft connected by gearing with a vertical shaft, *A* (Fig. 302), on which a cogged wheel, *B*, is mounted *eccentrically*. The cogs on *B* engage with similarly pitched cogs on a rack, *C*, which is in rigid connection with a crosshead, *F*. When space

permits, the crosshead *F* is fixed on to the rudder head directly; but generally space does not admit of sufficient movement of the rack, and the crosshead *F* is connected with the crosshead on top of the rudder by rods, *GG*; and *D* is simply a vertical shaft on which the rack *C* and the crosshead *F* are mounted. The curvature of the rack is obtained graphically, so that when each of the numerals, shown on the wheel and rack, come into position on the fore-and-aft line joining the two centres of the shafts *A* and *D*, the cogs are properly engaged.

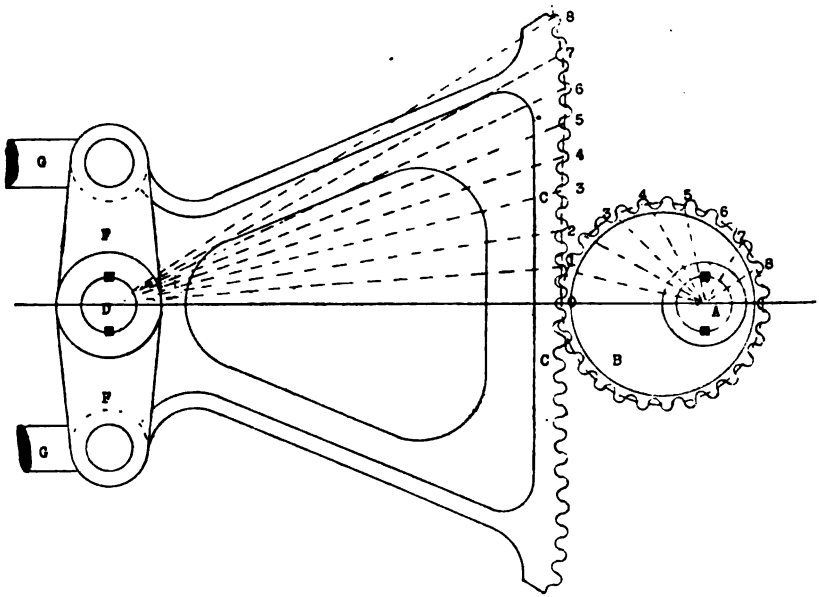


FIG. 302.—Harfield's Gearing.

The advantage of Harfield gearing is that the force or turning moment applied to overcome the resistance of the rudder is increased when the rudder is moved farther away from its middle position. Thus at the middle position the ratio of the force to the resistance is as *DO* is to *AO*, and in the extreme position the ratio is as *D*8 is to *A*8.

If a worm screw is used in the connection of the engine shaft with the shaft *A*, the gearing is self-holding; that is, if the shaft breaks at some point previous to the worm shaft, the gearing holds the rudder in one position. But if an ordinary series of bevel

wheels is used, without any worm screw, the gearing is not self-holding, and leaves the rudder free to move to and fro. Some arrangement is then sometimes fitted to hold the rudder in case of accident, and a rod working a piston in an oscillating cylinder is attached to one of the pins in the crosshead *F*. The cylinder is used as a buffer, and can be filled with water under pressure.

Wilson and Pirie's Gear.—In this gearing the pinion is fitted *concentric* with the shaft, the rack is circular (generally a quadrant),

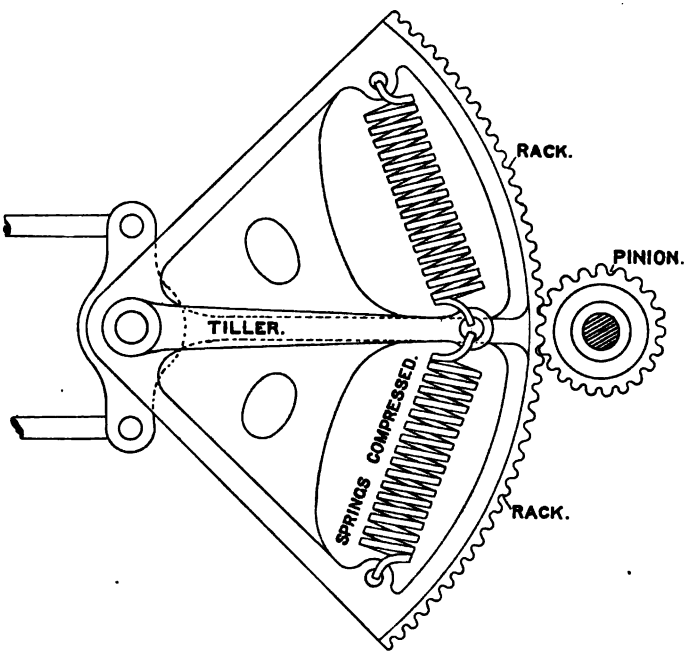


FIG. 303.—Wilson and Pirie's Rudder Gear.

and motion is imparted from the rack to the tiller through strong spiral springs.

The tiller is separate from the quadrant, but pivots about the same centre, and is connected, through a hole in the quadrant, with the rudder, or false rudder, head. The springs are always *under compression*, and take any shock caused by alteration of helm or motion in a sea-way. The gearing is thus compensated for back lash, and generally noiseless.

Muir's Hunting Gear.—In Fig. 304 the rod *A* is revolved by the steersman, and keyed to the toothed wheel *B* (23 teeth). *C* is

a smaller toothed wheel, or pinion (12 teeth), geared into *B* and *D* (46 teeth), and working loose on a pin fixed on *E*. *E* is a disc and shaft, free to revolve in its bearing, *F*, to which is rigidly attached a cam or eccentric sheave, *G*. Any movement of the cam operates on the differential valve and rod *H*. When the differential valve is moved it sets the engine in motion, revolving the worm screw *K*, which is keyed to the crank shaft of the engine. *K* is geared into the toothed wheel *M*, which, by means of gearing,

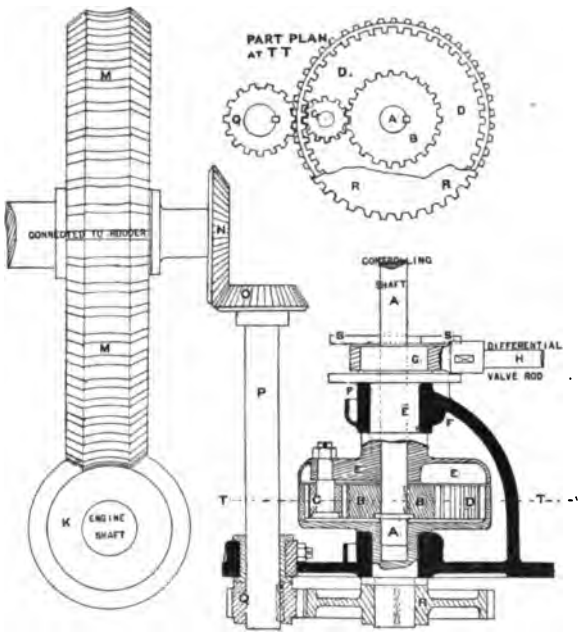


FIG. 304.—Muir's Hunting Gear.

acts on the rudder; at the same time *M* operates, through the mitre wheels *N* and *O* (36 and 24 teeth respectively), and the shafting *P*, the toothed wheel *Q* (17 teeth). *Q* gears into another wheel, *R* (51 teeth), which is keyed to *D*, and revolves with it.

The motion given to *D* is the opposite of that given to *B*, so that the pinion *C*, which moves with the cam, is returned to its shut-off position by the movement of the engine.

The cam *G* can be moved about a quarter of a revolution each way, and is then stopped by the plate *S*, which is cut and fitted for this purpose. Accordingly, if the engine does not follow the move-

ment of the steering wheel, the movement of the latter is limited by this plate. *S* also limits the movement of the differential valve.

If the friction of the controlling shafting is insufficient, and the steersman does not hold the wheel, it is just possible for *A* to be revolved by *D* through the medium of *C*.

Davis' Hunting and Steering Gear.—The engine is of the ordinary two-cylinder type, with two cranks set at right angles to each other, and the controlling gear, working a differential valve, sets the engine in motion. The engine can be applied to work either screw gearing, or (as shown in the figure) a wire rope or chain, or a pitch chain, or any other of the arrangements described.

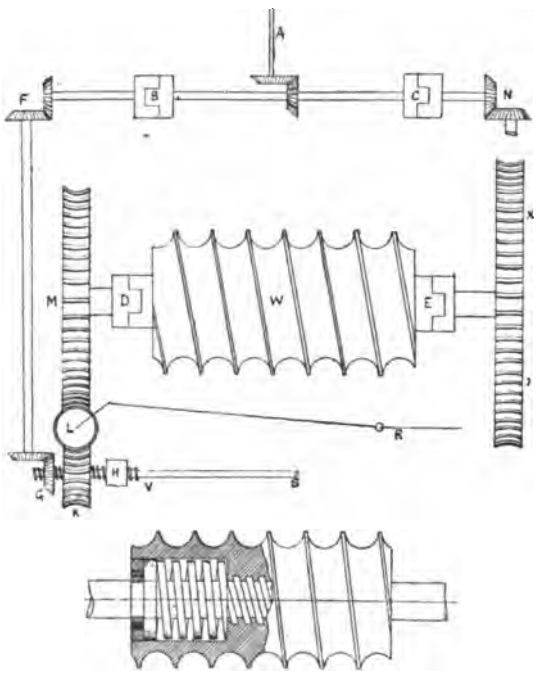


FIG. 305.—Davis' Hunting and Steering Gear.

The arrangement shown in Fig. 305 is used for chain gearing and in a small vessel. When steam gear is used, the clutches *C* and *E* are thrown out of gear and the clutches *B* and *D* are put in gear. The controlling shaft *A* is moved by the steering wheel, and moves the bevel wheels *F* and *G*. The bevel wheel *G* is fitted with a feather, projecting into a feather-way cut along the screwed spindle *VS*, so that when *G* rotates the spindle is rotated with it. The spindle *VS*, which is attached to the differential valve, being screwed through the centre of the wheel *K*, which is free to revolve, but not free to move along its axis, moves the valve to right or left, according to the direction of rotation imparted by the wheel *G*. The movement of the valve, setting the engine in motion, revolves the worm screw *L*, which gears into spur wheels, *K* and *M*. The

motion given by the engine to K moves the spindle in the opposite direction to that given by the controlling shaft, and, returning the valve to its central position, stops the engine. The wheel K and nut H are in one piece, and revolve together, but motion along their axis is prevented by a fixed bearing.

The engine driving the worm L rotates the spur wheel M, which is coupled by the clutch D to the barrel or drum W. A few turns of a wire rope or chain are wound round W, and the ends are led over loose pulleys at opposite sides of the tiller, to which they are attached. Any movement of the drum unwinds one end and winds up the other end of the rope, and thus moves the tiller. Stretching or tightening screws are generally used to keep the rope taut, and are generally used to connect the rope to the tiller. Other systems are also used for this purpose.

For steering by hand-power, the clutches B and D are thrown out of gear and the clutches C and E put in gear. The rotation of the controlling shaft A revolves N, which is connected with a worm screw (not shown), geared into the spur wheel XX, and thus rotates the drum W. The controlling shaft would not usually be strong enough for hand steering, and for large vessels a separate and stronger shafting is necessary.

Inside the drum W a *spring buffer* is sometimes fitted, as shown in the lower figure. The drum is secured to a shaft by a square-threaded screw, and a certain amount of axial movement of the drum is therefore allowed. At each end of the drum a concentric recess is formed, in which, and around the shaft, a spiral spring is inserted. The springs bear against collars within the recesses and fixed on the outward ends of the shaft. The compression of the spring by the movement of the shaft allows a certain range of elastic movement, and thus relieves any sudden strain on the rope or chain.

Capstan Engine and Gear.—The capstan engine and gear are used for getting up the anchor, and catting it if necessary, and for warping or hauling the ship from one position to another. The engine is usually of the two-cylinder type, with two cranks at right angles on the shaft, and it is controlled by a differential valve similarly to the steering engine.

In Fig. 306 an elevation and plan of the general arrangement of two engines, two cable holders or windlasses, and two capstans are shown, as fitted by Messrs. Napier Brothers in several large warships and mercantile vessels. In warships the engines and gearing

are fitted below the armour and deck protection; this entails longer spindles and control gearing shafting, but otherwise the arrangement is substantially the same as shown in the figure. *A* is the control wheel and shafting, which can be worked from the forecastle deck

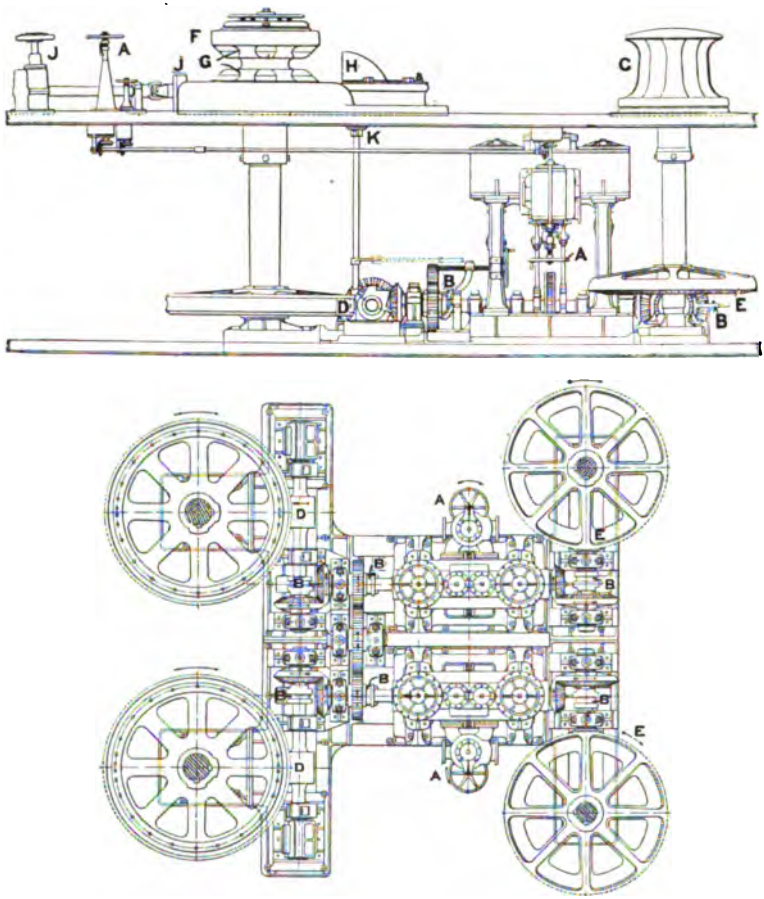


FIG. 306.—Napier's Capstan Engine and Gear.

or at the engine. By means of clutches and couplings, *B*, either engine can be used in connection with either cable holder, *F*, or capstan, *C*. The two capstans are used ordinarily for hauling or warping, and, where necessary, for catting. The two cable holders are used almost exclusively for weighing the anchor, and due to the worm screw *D*, instead of bevel pinion, *E*, gearing move more

slowly but exert greater force. The cable is passed half round the holder *F*, fitted with stops, *G*, to obtain a good grip, and then falls through the navel pipe *H* into the chain locker below. The cable holder can be held in a fixed position by a friction brake, *J*, of which there are several varieties, and the cable is held by the compressor *K*. The compressor is now usually made as a lever pivoted horizontally to one side, close up and below the navel pipe. When not in use, the cable is free to run through a hole (of the same diameter as the navel pipe) in the lever; by pulling the lever to one side, using a tackle for the purpose in large ships, the cable is "bowsed to" by closely nipping it between the compressor and the edge of the navel pipe. As additional securities, the cable is usually "bitted" and a "slip" fitted.

Relative Stress in Cable and Steam Pressure in Cylinders.—Of the indicated work done on the engine pistons, a reduction is caused respectively by the engine friction, friction in the bevel gearing (see Fig. 306), the worm screw gearing, the cable holder spindle, and of the cable moving over the deck, etc., and through the hawse pipe. For ordinary purposes of calculation, the respective efficiencies may be taken as, $\frac{4}{5}$, $\frac{4}{5}$, $\frac{1}{2}$, and $\frac{4}{5}$; and the overall efficiency—

$$E = \frac{4}{5} \times \frac{4}{5} \times \frac{1}{2} \times \frac{4}{5} = \frac{1}{4} \text{ about.}$$

The stress or pull in the cable just outside the hawse pipe is evidently of the same nature as that of a pull in a tow-rope (see Chapter XXIV.); and if

h = height through which the anchor is raised per revolution of the engine shaft, or the distance through which the ship is pulled—

$$\text{Stress in cable in } lb. = \frac{2\,P.L.A}{h} \times E \quad . \quad . \quad (1).$$

In the above formula for the same engine and gearing L, A, and h are constant qualities, and E is also approximately constant; and thus—

Stress in cable $\propto$ P (the pressure in the cylinders).

Or, in a simple case where the weight W of the anchor and cable is known, and the rate x (generally 25 feet per minute) of raising the anchor is specified—

$$\text{Effective H.P.} = \frac{\text{W lb.} \times x \text{ (feet per minute)}}{33{,}000},$$

$$\text{which } \text{„} = \text{I.H.P.} \times \text{E.}$$

$$\left.\begin{matrix}\text{Stress in cable} \\ \text{in lb.}\end{matrix}\right\} = \frac{2\,\text{P.L.A.N}}{\text{N} \times h} \times \text{E, from} \quad . \quad . \quad . \quad (1)$$

$$= \frac{\text{I.H.P.}}{\text{N} \times h} \times \text{E} \times 33{,}000, \text{ in which N} \times h = x \text{ evidently,}$$

$$= \frac{\text{I.H.P.} \times 33{,}000 \times \text{E}}{x}.$$

In actual practice the weight W includes an allowance for the holding force of the anchor when embedded in the sea bottom, and also an allowance for the frictional resistance of the cable up to the point of suspension to the cable holder. An overstress, or *strain*, is prevented by limiting the pressure in the engine cylinders, so that the power exerted by the engine shall not be in excess of that allowed for the cable stress.

EXAMPLE.—Find the I.H.P. of engine required for lifting a weight of 35 tons (equivalent to the cable stress) at the rate of 25 feet per minute.

$$\text{I.H.P.} = \frac{35 \times 2240 \times 25}{33{,}000 \times \text{E}} = \frac{60}{\text{E}} \text{ nearly.}$$

And if $\text{E} = \frac{1}{4}$, then I.H.P. of capstan engine

$= 240$ nearly, which is not uncommonly fitted in large battleships and cruisers.

It should be remembered that a capstan engine is fitted with a differential reversing valve and normal slide valves, and that the engine is almost entirely non-expansive working. The steam consumption per I.H.P. is therefore very large, and consequently a large boiler power is necessary for working the capstan and weighing anchor; and roughly, a boiler which will produce 1000 or more I.H.P. when supplying the main propelling engines, will barely supply a capstan or steering engine *working continuously* at 240 I.H.P.

CHAPTER XXVII

WATER-TIGHT, PUMPING, AND FIRE SYSTEMS

Water-tight Subdivision.—A ship of any importance is divided into a number of *water-tight compartments*, so that in the event of one or more being flooded by grounding or collision the ship may still remain afloat. These several compartments are separated by steel-plate bulkheads, strengthened as necessary by vertical steel beams of Channel, *H*, or *Z* section. The principal bulkheads extend across the whole transverse width of the ship and are known as *transverse* bulkheads, and by their means the ship is divided into nine or ten or more *main* water-tight *compartments*. Longitudinal bulkheads extend between the transverse bulkheads as necessary, and thus subdivide the main compartments into a large number of small compartments which assist, if required, in keeping the ship afloat. The largest compartments are usually the boiler rooms (three or four or more in large battleships, cruisers, and passenger steamers) and the engine rooms (two or more). Next in cubic capacity are the coal bunkers, which are separated from the engine and boiler rooms by longitudinal bulkheads, and cargo holds which may extend the whole transverse width between the inner or outer bottoms. There are other large compartments, such as the capstan engine, submerged torpedo, steering, and tiller rooms, besides a great number of smaller spaces for stores and ammunition. Within the outer bottom, and extending from the keel to somewhere about the water-line, there is another and inner bottom, and between these two inner and outer bottoms are a number of water-tight compartments, of which those near the keel are known colloquially as the *double bottoms*, and those near the water-line as the *wing compartments*. Thus, in case of collision say, about amidships and in wake of the boiler rooms, three bulkheads must be pierced (viz. the outer and inner side or bottom

plating, and the bunker bulkhead) before water gains access to the boiler room. By this system of subdivision a well-built steel ship is well protected from sinking after collision or grounding.

Water-tight Doors.—For access to the various compartments and for communication between them it is necessary to have doors which, when closed, make each compartment a water-tight space. These doors are of various types according to their position and importance.

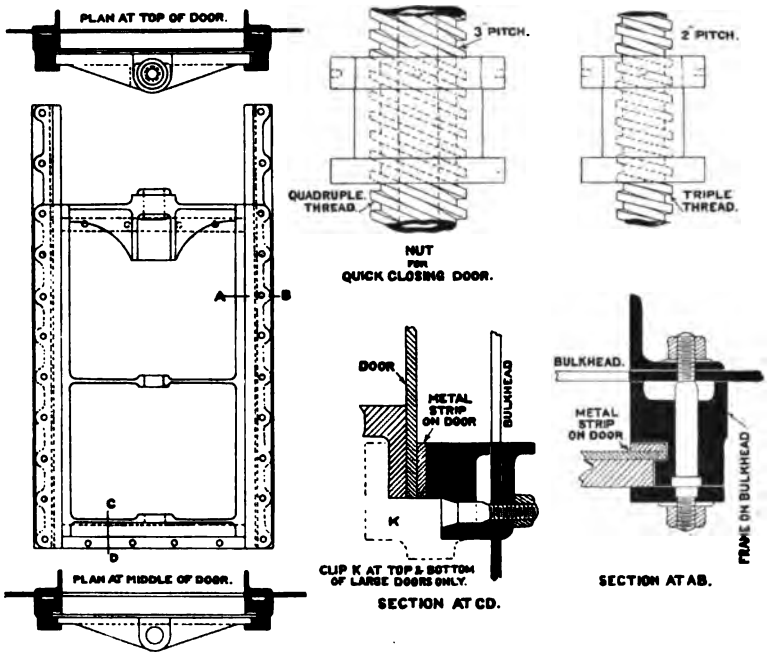


FIG. 307.—Vertical (Geared) Water-tight Door.

For the wing compartments, double bottoms, lower parts of the hold, forward and after (narrow) parts of the ship, and a few small store rooms, ordinary *manhole doors* (similar in character to those used on top of cylindrical boilers) secured by bolts and butterfly nuts are fitted as necessary, one or more to each compartment. These doors are only opened for cleaning, painting, or examination of the plating of the compartment, and are always closed and secured at sea.

For the main transverse and bunker bulkheads, where the height permits of their use, *vertical (geared) water-tight* doors are fitted, as shown in Fig. 307. All geared doors are connected through rods

and gearing with, and can thus be worked from, the main or upper deck, as well as from one or both sides of the bulkhead near the door. In recent warships the bulkheads between the several boiler and engine rooms are *solid*, *i.e.* no doors are fitted in them. Bunker doors are, however, necessary between the several bunkers and boiler rooms, so that the coal can be brought to the fires. In warships generally the bunker compartments are arranged in two tiers, upper and lower, with an armoured (sloping) deck between them. The lower transverse bulkheads are usually solid, but doors of the hinge-clip variety are usually fitted in the upper bunker transverse bulkheads. Between the upper and lower bunkers geared doors (armour thickness) are fitted, and special arrangements are made to prevent the openings from choking and thus not allowing the door to be closed. These doors are sometimes worked through rack and pinion gearing, as shown in Fig. 309.

The vertical sliding water-tight door consists of a flat plate strengthened by ribs on the back (as shown in Fig. 307), and it is faced with a continuous strip of brass or gun metal where it comes in contact with the frame to make a water-tight joint. The frame is bolted to the bulkhead and makes a permanent water-tight joint with it; on each side there is a vertical wedge-shape groove, tapered about $\frac{3}{16}$-inch per foot run, in which the door can be moved up or down. The door is raised and opened by means of a long screwed spindle working in a nut fixed in the top rib of the door. When the door is closed down, the wedge shape presses the metal face against the frame and completes the joint. It is necessary that these doors should be closed quickly, and, as shown in one of the small views, the spindle is sometimes made with a square three or four threaded screw; this quick-closing arrangement is correspondingly slow when opening, but this is a matter of no importance. The details of the ordinary construction are shown in the sketches.

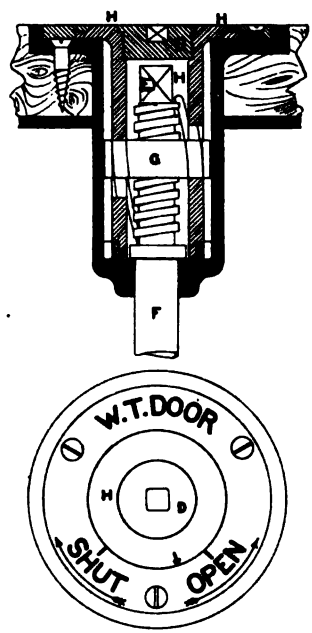


FIG. 308.—Deck Plate and Indicator.

Fig. 308 shows a vertical section and plan of a *deck plate and indicator* used in connection with the gearing for working a water-tight (geared) door. The cap *D* is first removed, and then a square socketed spanner is inserted and fitted over the end *E* of the gear

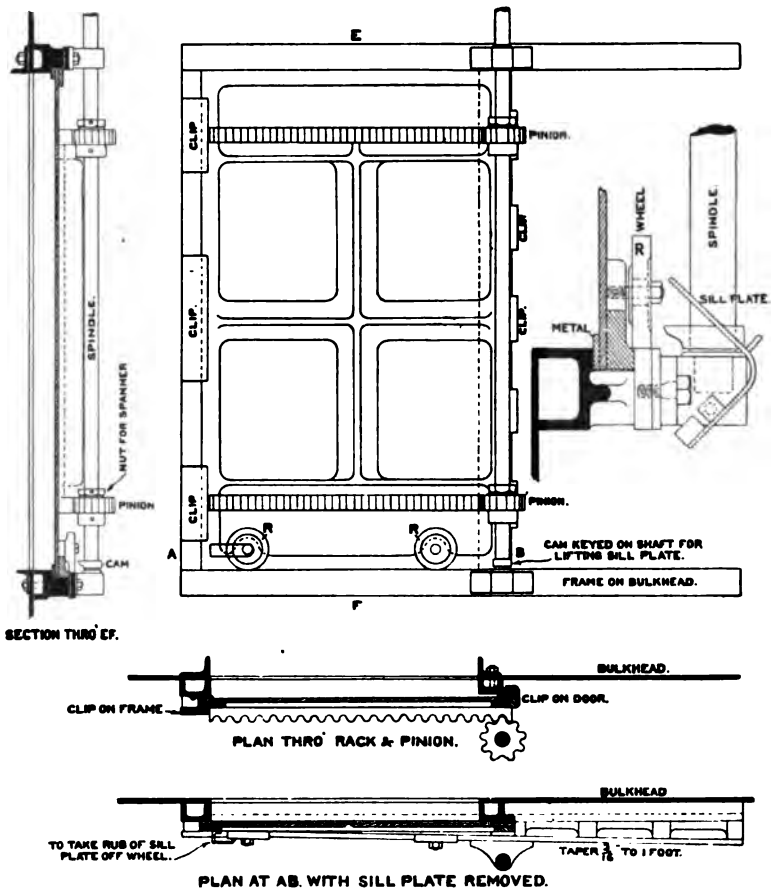


FIG. 309.—Horizontal (Geared) Water-tight Door.

spindle *F*. The rotation of the spindle raises or lowers a nut, *G*, which, being prevented by side lugs from rotating, rotates the indicator cylinder *H* by means of extensions of the lugs into nearly vertical grooves. On the top of *H*, and level with the deck, an arrow is engraved which points to the position of the door, whether open or shut.

The *horizontal sliding water-tight door* is of similar construction to the vertical variety, but, instead of a long screw, a rack and pinion gearing is used for opening and closing it, as shown in Fig. 309. The weight of the door is supported by rollers, R, and a space is generally left below the lower wedge-shaped groove to allow the dirt to fall clear and thus not block the passage of the door when closing it. Over the sill of the doorway a sill-plate is fitted which is pushed away automatically by the door when closing it. At the back edge of the door, clips are fitted which swing automatically into place and assist in holding the door closely against the frame.

Hinged water-tight doors are used in places of minor importance, and above the water-line particularly. The frame is riveted to the bulkhead, and a strip of india-rubber or sheet asbestos is fitted to the frame and forms a continuous face, against which one edge of an angle steel (riveted to the edge of the door) is pressed when the door is closed and clipped up. These clips are a weak and unsatisfactory fitting; they are liable to fracture under a sudden stress, and are usually incapable of maintaining water-tightness under a heavy pressure.

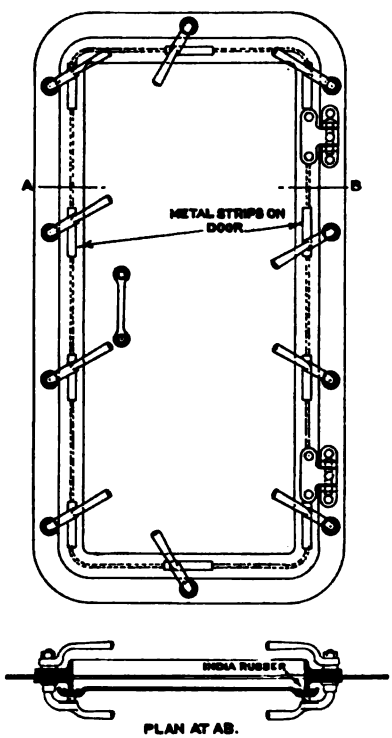


FIG. 310.—Hinge-clip Water-tight Door.

Below the water-line and over important deck openings geared sliding hatches, of similar construction to the horizontal (geared) door and to those used between the upper and lower bunkers, are fitted; but for deck openings above the water-line, especially in the armour decks, *armoured hatches* are fitted and hinged to the deck. These hatches are frequently very heavy, and differential chain tackles are used for opening or closing them. These hatches are not usually an entirely water-tight fitting; but clips are fitted, and

generally a good joint can be secured by pressing them down on soft packing or gasket.

For the purposes of drainage from one compartment to another *sluice valves* are sometimes fitted instead of ordinary stop valves. The sluice valve is a vertical (geared) door of small size (from 3 inches square upwards), and in important positions is generally fitted with an indicator plate so as to be workable from the main or upper deck. Although convenient for purposes of drainage, there can be no doubt that in many warships they are far too numerous, and the majority could be dispensed with. A suction pipe might be substituted in some instances with advantage, and if properly fitted with non-return valves, each compartment can easily be isolated; whereas drainage by means of the ordinary sluice valves entails flooding a compartment, to a more or less extent, which should not be flooded. Under certain circumstances of grounding or collision sluice valves become a dangerous fitting, and in one or two well-known instances disaster has been only narrowly averted.

In many of the older type of warships, sluice valves were commonly fitted in the ventilation trunks leading from one main transverse compartment to another. In such cases, similar dangers attend their use as in the case of drain sluices; but the danger is not so pronounced, because the ventilation valves are usually near the tops of the compartments, and there may be a better opportunity for closing them in emergency. In all recent ships, both mercantile and naval, each main transverse compartment is usually self-contained in regard to ventilation, and ventilation trunks are independent of those in other main compartments. This arrangement not only increases the safety of the ship but also improves the ventilation. Since attention was first called to these important matters of water-tight fittings and ventilation, in the first edition of this work in 1896, considerable improvement has been made; and there only remains to add the caution of not relying on any pumping arrangement, however efficient it may be, but rather to ensure every bulkhead being really water-tight and efficient enough to keep the ship afloat.

Stone-Lloyd Water-tight Door Control System.—In this arrangement the doors are operated by hydraulic power supplied by pumps always at work. The hydraulic pressure pipes, as fitted in the *Mauretania*, are in connection with a screw-down valve, the operation of which from a pedestal on the bridge, by means of steel wires and

chains, admits pressure into a subsidiary line of piping and closes all the doors simultaneously. When the clutch-gear for closing the doors is moved to admit pressure, a large gong is rung at each bulkhead door station and warns every one of the impending closing of communication. A special arrangement is made so that a man shut into any compartment can liberate himself through the door, but the operating lever must be held continuously or the door recloses automatically. The door closes in all cases slowly, and the power is sufficient to break up any obstructions.

Power available for pumping Water from the Ship.—In a battleship, the following engines are available for pumping water from the bilges:—

1. Four main circulating pumps, with a total capacity of from 6000 to 10,000 tons per hour. These pumps draw water from the engine-room bilges only, but pipes are led independently from each boiler-room bilge to drain water into either engine room. In draining water from the foremost boiler room into either engine room, the intermediate boiler rooms are not necessarily flooded.

2. Two fire pumps, with a total capacity of about 200 tons per hour.

3. Two bilge pumps, with a total capacity of about 200 tons per hour. (For small vessels, bilge ejectors are fitted.)

4. Several hand pumps of the Downton type. These pumps are fitted to draw from the lower part, or the bilge of each main transverse compartment. The leads of pipes are very systematically arranged, but the efficiency as suction pumps is very low, and they are seldom useful for suction depths of more than 20 feet. To render them more useful, the pump chambers and boxes should not be more than 15 feet above the lowest parts of the bilge suction boxes.

5. Special ballast pumps.

Fire and Bilge Pumps.—Each pump is driven by a separate engine. They are usually of the same size, with interchangeable working parts, and fulfil the same purpose. It is usual to keep the fire engines only for use in pumping on deck, so that only clean water shall pass through them, but in cases of urgency they are used as bilge pumps. In recent ships the steam and water pistons are fitted to the same piston rod, and no gearing or crank shaft is used. Independent suction pipes are led to each pump from each of the machinery and boiler spaces, from the sea, and from

the main drainage suction system, in connection with the various compartments of the ship. Each pump can deliver either into the fire-main or overboard. The pump valves are frequently made of vulcanised india-rubber; but sometimes flat metallic valves of the Kinghorn type are used, and are generally more reliable.

Bilge Ejector.—For small vessels the usual fire and bilge engines are too heavy, and can only be connected with the various compartments by long and comparatively heavy pipes, and therefore a bilge ejector is generally fitted to each compartment. About ten to fifteen such ejectors are fitted in a torpedo-boat destroyer, and each is capable of ejecting about 30 to 60 tons of water from the bilge per hour.

A bilge ejector is shown in Fig. 311. The top pipe is connected with the hand pump, and must be shut off when the ejector is working. The bottom pipe is led down to the lowest part of the bilge, and the end is left open and covered by a strainer. Steam is admitted through the small pipe, valve, and nozzle shown, and the mixed steam and water is blown overboard through the cock and pipe to the right. If the water is not immediately drawn up, when the discharge cock is opened and a little steam turned on, the apparatus can generally be started by keeping the cock closed and turning on steam gently. Condensation in the suction pipe (increasing the nozzle energy, see Chap. XXXIV.) draws up some water, and a sudden opening of the discharge cock generally sets up a discharge, when the steam valve can be opened gradually until a full discharge is obtained. Within the limits of effectual operation, the ejector should be fitted as close to the bottom of the bilge as possible, and as no suction is then exerted, the certainty of

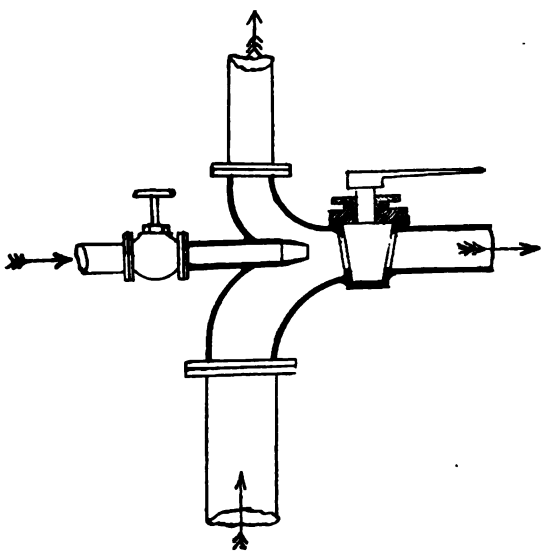


FIG. 311.—Bilge Ejector.

action is increased; but it is necessary to keep the ejector at a fair height above the bottom of the bilge for easy manipulation when the bilge is flooded, and therefore they are generally fitted above the water-line. The efficiency of the ejector depends very largely on the internal shape of the nozzle.

The objection to the use of ejectors is the loss of fresh feed water, which has to be made up, because it cannot be returned to the boiler. There is, however, no mechanism to get out of order; the dirt and small ashes are passed overboard without stopping its working, and the space and weight required are very small.

In some of the older and larger ships the ejector is fitted with a series of nozzles, and in the bilge without a suction pipe. The steam valve can be worked by gearing, from some point well above the bilge. The momentum of the steam will frequently raise the water 30 feet or more, but the consumption of steam is very great, and, of course, the fresh water is wasted.

Latrine Pump.—A small engine, of the same type as the fire engine, is sometimes fitted in larger ships for pumping through the fire-main and flushing the latrines. It is not usually available for any other purpose.

Quantity of Water entering a Vessel through a Hole.—For this calculation the following figures are useful:—

1 ton of sea-water occupies 35 cubic feet.

1 cubic foot of sea-water weighs 64 lb.

1 gallon of fresh water weighs 10 lb., and sea-water weighs about 3 per cent more.

1 ton of fresh water contains 224 gallons.

The velocity with which water will enter a ship depends on the *head*, or depth below the water-line. Call this H; then—

$$\text{The velocity} = \sqrt{2 \cdot g \cdot H}$$

where $g = 32$, approximately.

$$\text{Therefore velocity} = 8 \cdot \sqrt{H} \text{ in feet per second.}$$

From which, if A = area of hole in square feet,

$$\text{Tons entering per hour} = 820 \cdot A \cdot \sqrt{H} \text{ about.}$$

By simple calculation, the quantities in tons which will enter a ship through a hole of 1 square foot in area are as follows:—

Depth of Hole below Water-Line.	Tons of Water entering per Hour.	Remarks.
1 foot	820	For each 1 sq. ft. of area of hole
4 feet	1640	
9 ,,	2460	
16 ,,	3280	
25 ,,	4100	

By comparison with the pumping power, it is evident that only the leakage through a very moderate-sized hole can be kept under, and that the means of keeping a ship afloat depends principally on the water-tightness of the various compartments. It is hardly necessary to emphasise the immediate necessity of localising any leakage and isolating the compartment affected.

Fire Arrangements.—The fire and bilge pumps are available for the purpose of putting out a fire, and in addition, each Downton pump can be connected with the common delivery pipe, known as the *fire-main*. The fire-main run of piping is situated below the armoured deck or water-line, and runs fore and aft nearly the whole length of the ship. There are branches with valves and fittings for hose connections in the various compartments, and in each main transverse water-tight compartment a branch, known as the *rising-main*, leads up to the compartments above the armoured deck and water-line. A valve is fitted close to the fire-main, so that each rising-main can be shut off in case of injury to it, and at certain bulkheads shut-off valves are fitted to allow various sections of the main run of pipe to be shut off and repaired. The forward rising-main is commonly used for washing down the cable when getting up anchor, and is termed No. 1 rising-main, and the remainder are numbered consecutively—No. 2, No. 3, etc.

In warships, valves and cocks connected directly with the sea are fitted for *flooding the magazines*, etc., when required. The flooding valve for each individual magazine is locked, so that it cannot be tampered with. One main sea connection is generally provided for each group of magazines, and branches are led from it, as necessary, to the locked flood valves.

The sea connections throughout the ship are now generally ordinary stop valves, made entirely of special brass. A short pipe, where necessary, extends between the valve (which is next the inner skin) and the outer bottom. On the outside around each opening a

zinc protector is fitted. The zinc does not prevent galvanic action, but is more susceptible than either the steel bottom plating or the brass valve, and attracts to itself nearly all corrosive and galvanic action. These zinc protectors are usually renewed on occasions of docking, or about once a year. Similarly, protection from corrosion and galvanic action is fitted in the region of the stern bearings and screw propellers.

CHAPTER XXVIII

AIR COMPRESSING AND HYDRAULIC MACHINERY

Air Compressing Machinery

In the Royal Navy, compressed air is used for three principal purposes:—

1. For supplying air to the gas mixers fitted above the fires in Belleville, Niclausse, and other boilers; working pressure usually not exceeding 30 lb. per sq. inch.

2. For removing the soot and small scoriae from the outer surfaces of the tubes of boilers of the Babcock and Wilcox and other types, where manual appliances cannot be quickly applied; working pressure usually not exceeding 100 lb. per sq. inch.

3. As the motive energy of torpedoes of the Whitehead type; working pressure up to 2250 or 2500 lb. per sq. inch. And for forcing the torpedo from the directing tube (or gun); working pressure up to 750 lb. per sq. inch supplied from a reservoir.

Blowers for Gas Mixers.—The non-cooled compressor used for supplying the gas mixers is merely a common plunger pump, which takes its suction from the compartment in which it is placed. Being fitted in the upper part of the boiler room, it assists in the ventilation by drawing in hot air, temperature 120° to 140° F., and supplies the furnaces with air at a much higher temperature, frequently 500° to 600° F. The work done in compressing the air is thus partly returned to the furnace, and the hotter air assists in producing smokeless combustion.

Being subject to a high temperature, the delivery pipes, etc., must be made of metals of higher melting point than tin, lead, or zinc; and the flanges should be brazed on and not soldered. Bellows expansion pieces are necessary for straight runs of piping unless expansion is provided for in other ways.

The working pressures for various rates of combustion are given in Chapter X. The blower is also useful for removing the dust and loose lime scale from the interior of Belleville and other boiler tubes after cleaning and before washing out.

Blower for Sweeping Tubes. —Fig. 312 shows a Brotherhood compressor used for moderate air pressures of 80 to 250 lb. per square inch. The steam cylinder *A* is at the top, and the crank shaft and closed chamber, *B*, at the bottom, with the compressor cylinder *C* between. The crank shaft serves to drive the slide valve for controlling the supply and exhaust of the steam to the cylinder *A*, and also to limit the piston stroke and ensure its completion. The crank chamber is closed, and the crank and connecting-rod bearings are lubricated by oil compressed and forced through them by the oil pump shown. This pump is of the plunger type, without valves, and of similar construction to that used for forced lubrication of the electric light engine shown in Fig. 323. A small pump is also fitted, in connection with the slide valve rod, to circulate water through the compressor cylinder jacket; by this means the air is delivered at nearly atmospheric temperature. The inlet water delivered by the pump generally enters the water-jacket space *D* at the bottom end of the compressor cylinder *C*; then passes upwards through the barrel jacket *E* into the top-end jacket *F* and thence to the outlet. Air from the surrounding atmosphere enters through the air inlet and inlet valves; the compressor is a double acting pump; and the compressed air passes away through the air discharge. The inlet and outlet valves are generally made of thin gun metal and

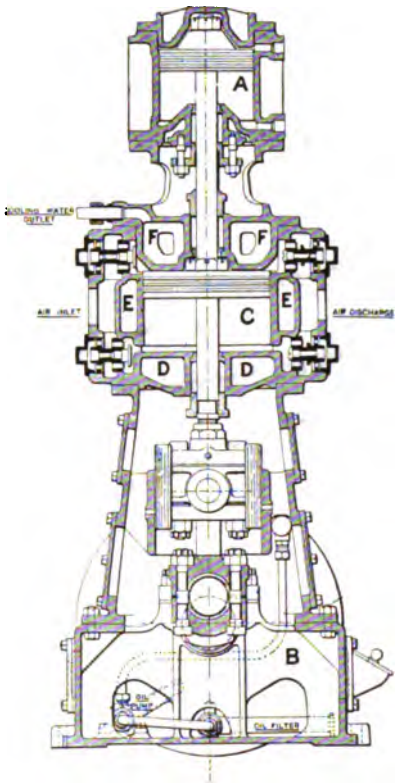


FIG. 312.—Air Compression, Single-stage Type.

have a limited lift; they can be examined and renewed without removing any part of the remaining fittings, and the arrangement is a very convenient and easily accessible one.

Multiple-stage Air Compressors.—As already explained in Chapter XXIII., it is necessary when compressing air to high pressures to not only water-cool the compressor cylinders, but also to cool the air between the various stages of compression; otherwise the energy expended in doing the work of compression would be unduly great and the efficiency unnecessarily low. Usually three or four stages of compression are embodied in the apparatus; and the higher the pressure attained the smaller the diameter or stroke-volume of the compression cylinder.

In *Brotherhood's four-stage air compressor*, Fig. 313, there are two steam cylinders, *AA*, and two compression cylinders, *C* and *D*, side by side; the pistons and plungers are all attached to a single cross-beam, *E*; and *E* is connected by double connecting rods with a crank shaft, *F*, which in its turn operates the steam slide valves through small cranks, *G* and *G*. A water circulating pump, *H*, of the plunger type is also attached to the crossbeam, and is worked through its means. The steam cylinders are of the usual kind and are fitted with cylindrical slide valves, one of which is shown in section in the left-hand figure.

Each compression cylinder is utilised for two stages of compression; the lower pressure above and the higher pressure below the pistons respectively. Air from the atmosphere, with a little distilled water and sperm oil continuously supplied, enters through *J* and the inlet valve *K* above the left hand (and larger) plunger on the down-stroke, and fills the cylinder. On the up-stroke the valve closes, and the air is compressed and passes through a non-return valve and the first-stage intercooler coils *L* into the space *M* below the piston. On the next down-stroke the air is still further compressed, passes through a non-return and the second-stage intercooler coils, *N*, into the space *above* the smaller right-hand compressor piston.

In the second compressor cylinder similar operations are continued, and finally the compressed air, now at its highest pressure, passes through the fourth-stage cooler *P* into a separator. This separator drains the water away automatically, or it can be drained by hand, and the air is then discharged into a reservoir or torpedo as required. With very high pressures the removal of the water is attended with considerable difficulty, because the density becomes very great.

The circulating cooling water passes first through the casing containing the first-stage intercooler, and thence through the space surrounding the compression cylinders and containing the second, third, and fourth cooling coils, to the discharge outlet.

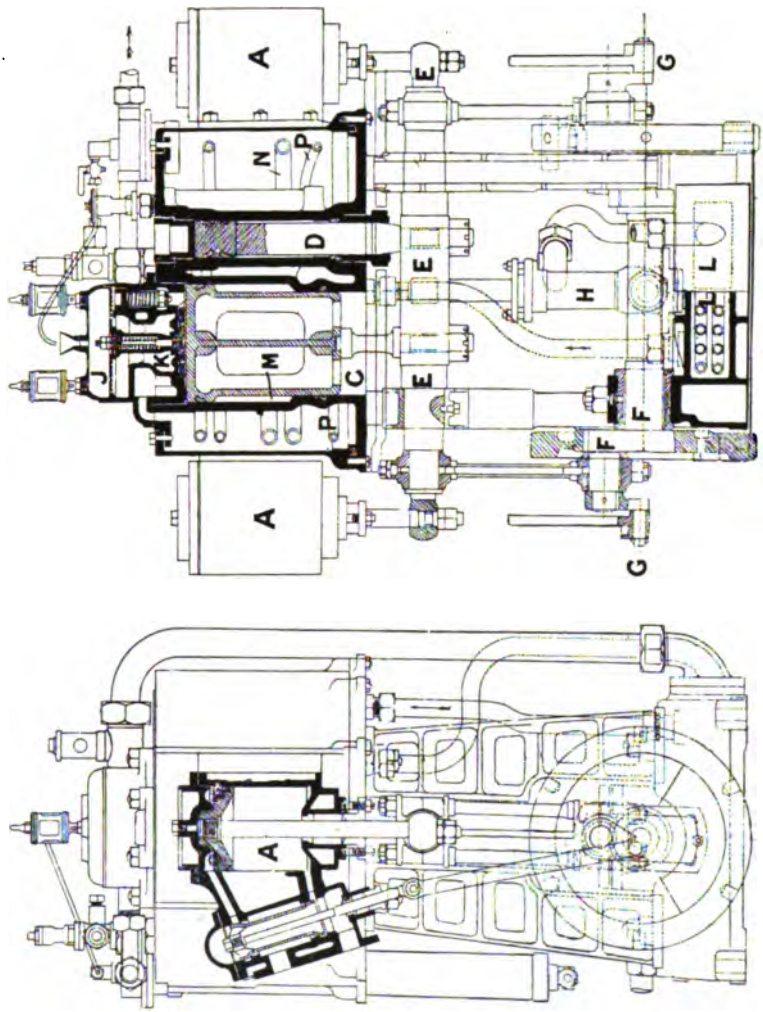


FIG. 313.—Brotherhood Four-stage Air Compressor.

Another type of four-stage compressor, made by Belliss and Morcom, is shown in Fig. 314, and a further description is hardly necessary. The various connections are clearly shown, and the working stages can be easily followed. The Belliss method of fitting the leathers is

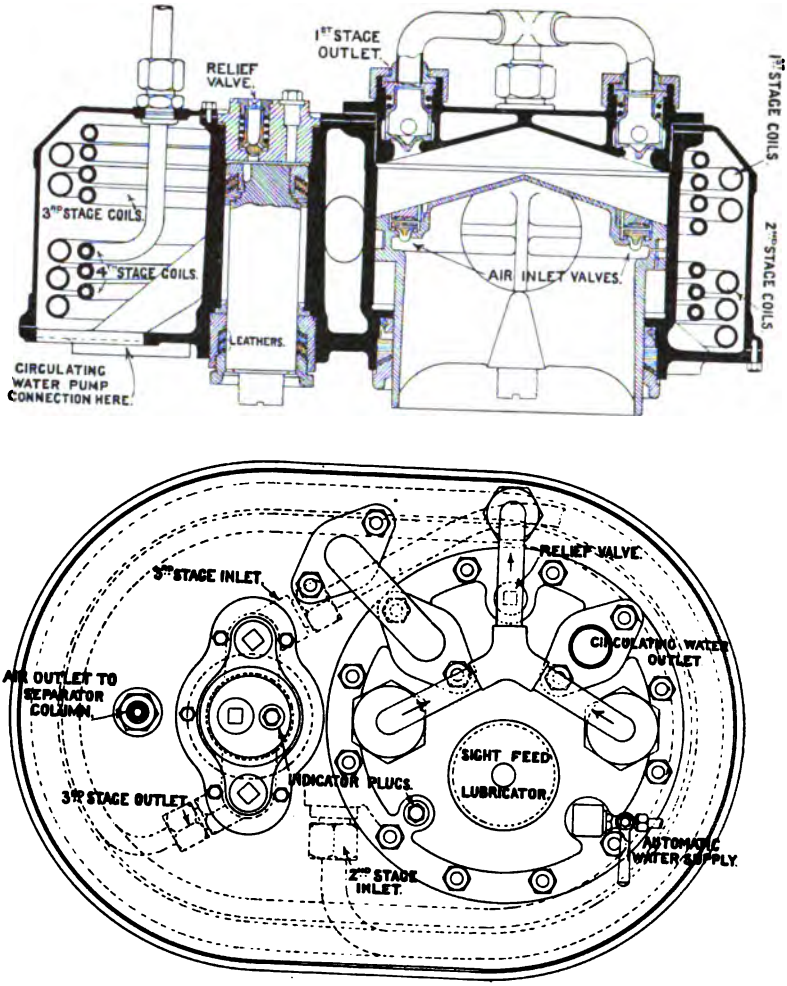


FIG. 314.—Belliss Four-stage Air Compressor.

FIG. 314A.—Belliss Packing Leathers.

shown in Fig. 314A, which also shows an enlargement of the packing gland and part of the small plunger rod of the H.P. stage of Fig. 314.

Hydraulic Machinery

Hydraulic (water) pressure is used in warships principally for loading and training heavy guns and turrets, for transporting and hoisting ammunition and projectiles to the turrets and guns, and for hoisting and lowering heavy boats and weights. In the mercantile marine it is sometimes used for loading and unloading cargo. In a few cases it has been applied for steering the ship, but no economical advantage is gained by its adoption in any of the mechanisms used for this purpose.

Until recently the engineer officer was entirely responsible for all hydraulic motors; but although he still deals with all important defects that may arise, and all repairs are naturally effected by the engineering staff both on board ship and in the dockyards, the Admiralty (very wisely, in the author's opinion) have to a large extent relieved him of the general routine duties, which are now carried out by the gunnery officer, who applies the motors to his special requirements. The motive power, being steam-driven, is of course supplied by the engineering staff, who are directly responsible for all hydraulic motors, such as boat hoists, which do not pertain to the guns. The Gunnery and Hydraulic Manuals deal very fully with all gunnery details, and consequently it is only necessary to herein describe those parts for which the engineer is directly responsible.

Hydraulic Pumping (Power) Engine.—The hydraulic motors derive their power from hydraulic pressure produced by one or more steam-driven pumps. The engine is usually a two-stage compound-tandem steam engine with the H.P. cylinder in front of the L.P. cylinder. The steam and pump cylinders are generally horizontally arranged, and the H.P. piston rod is directly coupled to a crosshead, to which the twin piston rods of the L.P. piston are also attached. The pump rod is also attached to this crosshead and is in the same axial line with the H.P. piston rod. Two such engines as above described are fitted side by side and coupled to the same shaft, with cranks at right angles, by the usual connecting rods which connect the crosshead and crank. The steam engine requires no special description, and is easily understood when seen.

One of the pair of pumps is shown in section in Fig. 315, and

the essential point in its construction is that the sectional area of the pump rod is made exactly one-half that of the plunger. The plunger is fitted with two U or L shaped cupped leathers (Fig. 315A), and is entirely water-tight for one direction of motion only: from right to left, as shown in the figure. The action is as follows: during the stroke, from right to left, water is drawn in through the suction

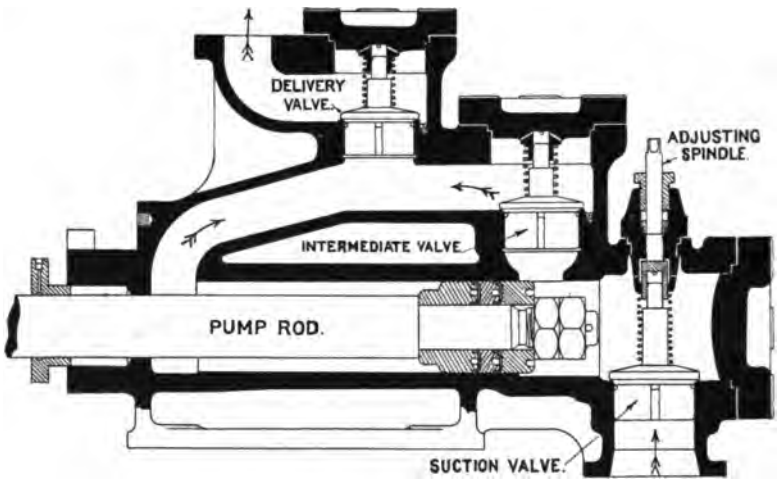


FIG. 315.—Hydraulic Power Pump.

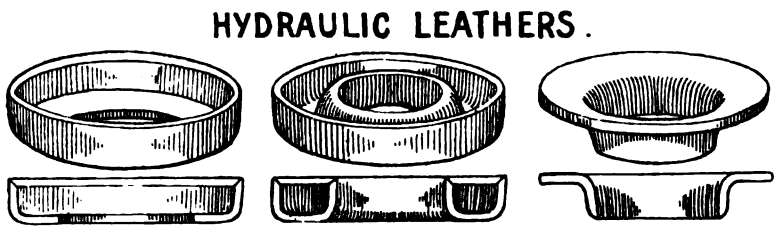


FIG. 315A.—Hydraulic Packing Leathers.

valve and fills the space behind the plunger and below the intermediate valve; during the inward stroke, from left to right, the suction valve closes automatically and the water is forced through the intermediate valve and fills the space around the pump rod and below the delivery valve. The pump chambers being now entirely filled with water, the plunger moves from right to left and twice as much water is drawn in through the suction valve as is delivered through the delivery valve; from left to right the plunger forces twice as much water through the intermediate valve as is required

to fill the space around the rod. There is thus an almost continuous and steady delivery of hydraulic pressure during both the forward and return strokes, although the suction is only open during one of them. Very important matters in the design of hydraulic machinery are: the avoidance of air pockets in pump chambers and pipes; a fair run of piping without abrupt changes in direction or sectional area of the moving water; and a steady and continuous flow without change of momentum and consequent shock and loss of energy.

Hydraulic Pressure Control and Governor.—Referring to Fig. 316, the crank shaft rotates the *speed governor shaft* by means of a pair of bevel wheels, B and C; the governor shaft is revolved at about two and a half times the rate of revolution of the crank shaft. When the engine speed is excessive, such as may be caused by a burst pressure main pipe, or by too many hydraulic motors at work at the same instant and thus developing an excessive power, the speed governor comes into play. The weights D tend to be thrown by centrifugal force away from the shaft, and this force, overcoming the springs E, carries the hollow spindle F to the right. F by means of a sleeve coupling acts on the lever GHJ, which is pivoted at H, and forces J to the left. J carries the rod KL with it and operates on the bent lever LMN pivoted at M, which, pulling the valve rod N downwards, tends to close the throttle-valve and reduce the steam-supply to the engine.

A nearly *constant pressure* is maintained by means of the *hydraulic governor*. Pressure from the main is admitted through a pipe, P, to the under side of a hydraulic ram, Q, which is connected by a crosshead at the top end with two springs in compression. When the hydraulic pressure exceeds that for which the springs are set, the ram Q rises and carries with it the lever RST, pivoted at R, and thus raises the servo-motor or auxiliary valve attached to the spindle T; pressure is thus admitted below the plunger or ram W, which is forced upwards and carries with it the connecting rod WX (not shown) and lever XKJ, which pivots on J; K is thus moved to the left, and tends to close the steam throttle-valve and reduce the speed of the engine.

The movement of the ram W corresponds exactly to that of the servo-motor valve on T, and when the hydraulic pressure is too low, the ram Q lowers and carries the servo-motor with it; the pressure escapes from below the ram W, through the exhaust cavity in the servo-motor and a central hole in the ram, into an exhaust space which is in connection with an exhaust return pipe. The

position of the nut N on the throttle-valve spindle is adjustable, as are also the hydraulic spring compression and length of the rod KL.

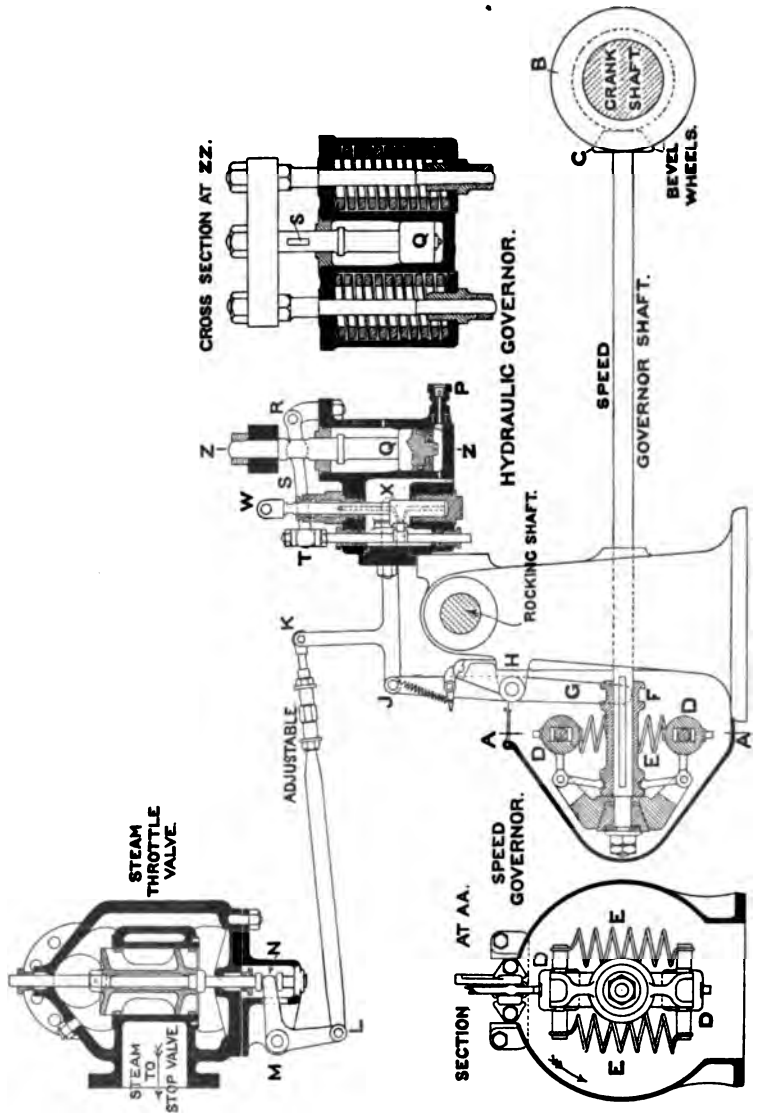


Fig. 316.—Hydraulic Governor.

Hydraulic Gun-training Engine.—Fig. 317 shows a type of hydraulic engine used for training the barbettes or turrets in which heavy guns are mounted, and Fig. 318 shows a section through one

of the slide valves *F* and *engine working* (differential reversing) *valve, H*, of the same engine.

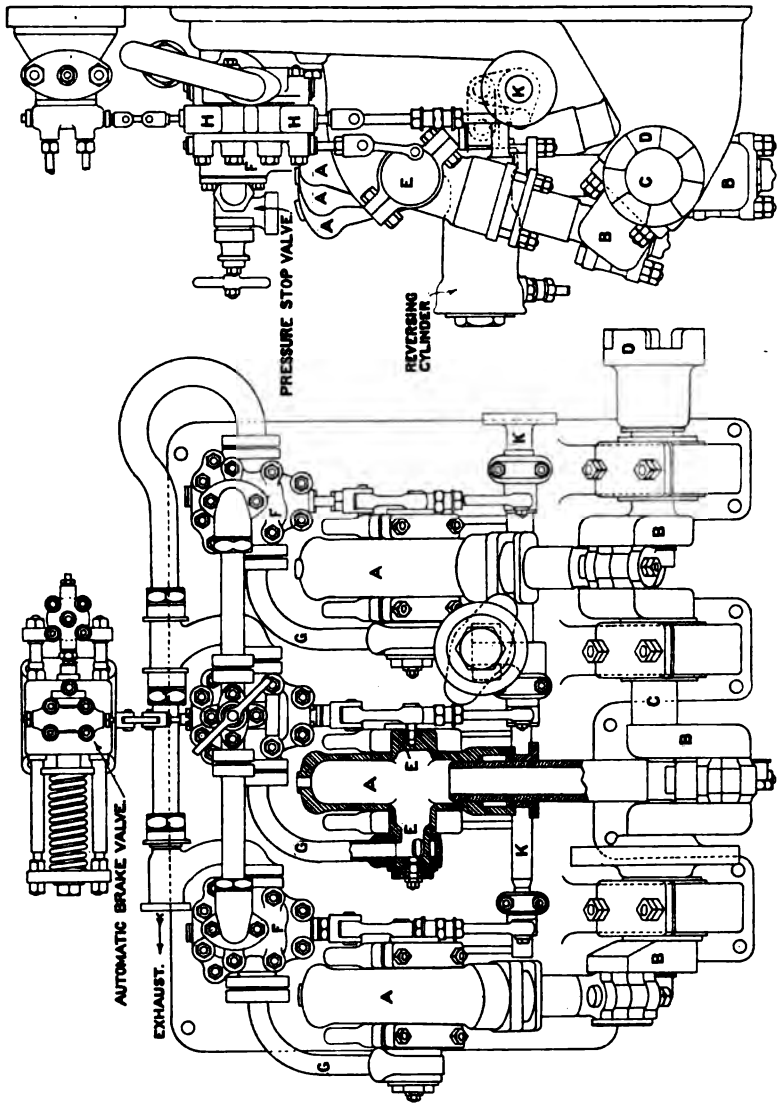


FIG. 317.—Hydraulic Gun-training Engine.

The engine consists of three single-acting oscillating cylinders, *A* (the middle one of which is shown in section in Fig. 317), mounted side by side on trunnions, *E*, and acting on three cranks, *B*, set at

equal angles (120°) on the same shaft *C* which is connected by a coupling, *D*, and gearing (not shown) with a rack on the turret wall. In the older ships, the training engine is situated outside the turret; but in the more recent ones the engine is fitted in a turn-table below the turret, and moves with the turn-table and turret which are rigidly connected together. Usually two engines are fitted for each turret, either one of which can be coupled or uncoupled as required. Each cylinder is connected, through one of its trunnions, *E*, and a pipe, *G*, with a slide-valve, *F*, and working (differential) valve, *H*; and by means of a reversing lever working a servo-motor valve in connection with the reversing cylinder, the engine can be reversed or stopped as required. The weigh shaft, or reversing shaft, *K*, is operated by the reversing cylinder, and is

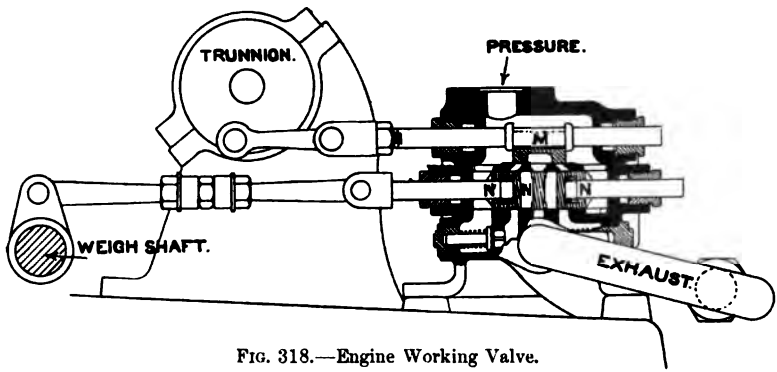


FIG. 318.—Engine Working Valve.

coupled by levers and connecting rods with the valve *H*. The action is precisely similar for each cylinder and set of valves, *F* and *H*; but each cylinder, being single acting, is in turn opened to exhaust, and the motion of the shaft is continued by the force imparted to it from the other two cylinders. Should the pressure in the system fall below a safe working limit, the Brake valve comes automatically into action and, overcoming the action of the reversing cylinder, centres the valve *H*, and by thus blocking the exhaust, prevents any further movement of the engine. The Brake valve is particularly necessary for boat and other hoisting engines when of the hydraulic type above described.

In Figs. 317 and 318 the letters *F* and *H* correspond to letters *M* and *N* respectively.

Referring to Fig. 318, the valve *M* is shown in the middle of its stroke, and its corresponding plunger would then be at one end

of its stroke. The weigh shaft is in the position corresponding to *stop*, and consequently the valve *N* is in the middle of its stroke with both pressure and exhaust ports closed. Suppose *N* is moved to the right, then no motion will follow, because the valve *M* covers both ports in the working face interposed between *M* and *N*. But the movement of the weigh shaft brings one of the other cylinders into play and sets the engine in motion, and moves *M* either to right or left as the case may be. Suppose *M* moves to the left; then pressure, passing through the false face and the right-hand port in *N* into the right-hand port below *N*, which is in connection with the cylinder, pushes the plunger outwards. Next, suppose *N* is moved to the left and *M* moves to the left, as before, under the influence of the other cylinders; then the exhaust cavity in the valve *M* connects the left-hand port below *N* with the exhaust, and, allowing the plunger to move back into its cylinder on the return stroke, the direction of motion of the engine is reversed.

The *Bollard engine* (hydraulic) acts on exactly the same principle as the engine above described, and the details are similar, but the cylinders are differently arranged. This arrangement corresponds to the three-cylinder Brotherhood engine generally found in torpedoes. All three plungers act on the same crank, and the cylinders are fixed at equal angles (120°) about the shaft. This type is generally used for small powers, hoisting ammunition, etc., and is more compact than that shown in Fig. 317.

The mechanical efficiency of a hydraulic engine, in combination with loss of efficiency in transmission, is low and generally less than that of electric power applied to the same purpose. Notwithstanding this low efficiency, there are many advocates of hydraulic power for working guns and hoisting ammunition in the British Navy; but in other navies the electric system is becoming increasingly adopted, although not entirely successfully. It should be remembered, however, that electricity for working heavy guns is comparatively in its infancy, and that during a period of over thirty years the hydraulic system has been steadily improved. Hydraulic power is heavier than electric, but it adapts itself more readily to variations of speed and load, and with men better trained in its use than with electricity repairs can probably be more readily made. Apart from these reasons, it is well to remember that electric power is cheaper, and of less weight, and occupies less space than hydraulic power, and in well-trained hands at least equally effective.

CHAPTER XXIX

REFRIGERATING MACHINERY

THERE are three principal systems of refrigeration in general use :—

1. The ammonia, or NH_3, compression system.
2. The carbonic anhydride, or CO_2, compression system.
3. The compressed-air system.

The objects of a refrigerator are to reduce the temperature in a cold chamber, and to maintain the lower temperature for long periods. A temperature of about 20° Fahr. is required for preserving meat, and not greater than 35° for preserving vegetables.

Principle of Refrigeration.—If gas is generated from a liquid in a closed vessel, the pressure continues to rise as more gas is generated, because there is only about the same volumetric space to fill. As the pressure rises, the temperature at which the liquid boils continues to rise also, at a rate accurately determined by experiment. If we take a certain quantity of this gas, at a certain pressure, and still further compress it without adding any more heat, some of the gas returns to the liquid state, because it is then compressed above the pressure corresponding to its boiling temperature under natural conditions; if the pressure be sufficiently increased and heat is also taken from it, the whole of the gas liquefies. This property is common to all gases, and is true of both ammonia and carbonic anhydride (carbonic acid gas).

Ammonia System.—Ammonia under the normal atmospheric pressure and temperature exists as a gas, but if compressed to about 105 lb. per square inch (or about 7 atmospheres), it becomes a liquid. In other words, the boiling-point of ammonia, under a pressure of 7 atmospheres, is about 60°, while under a pressure of

1 atmosphere it is about -38° Fahr. If the liquid can be reduced to a temperature of about -110° Fahr., it freezes and becomes solid.

The general system of ammonia refrigeration is shown in Fig. 319, but the sketch does not represent any particular plant. The gas is first compressed to about a pressure of 100 to 170 lb. per square inch, according to the temperature of the surroundings and of the gas-supply. The compressed gas is then conveyed through an oil separator, *S*, to a cooling condenser, where, the compression having already raised the boiling-point, the reduction in temperature converts the ammonia gas to a liquid. The liquid

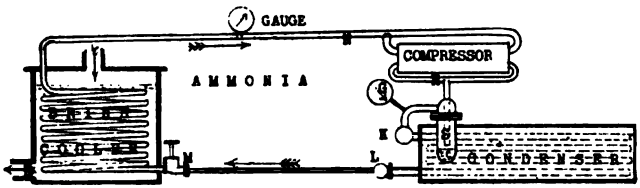
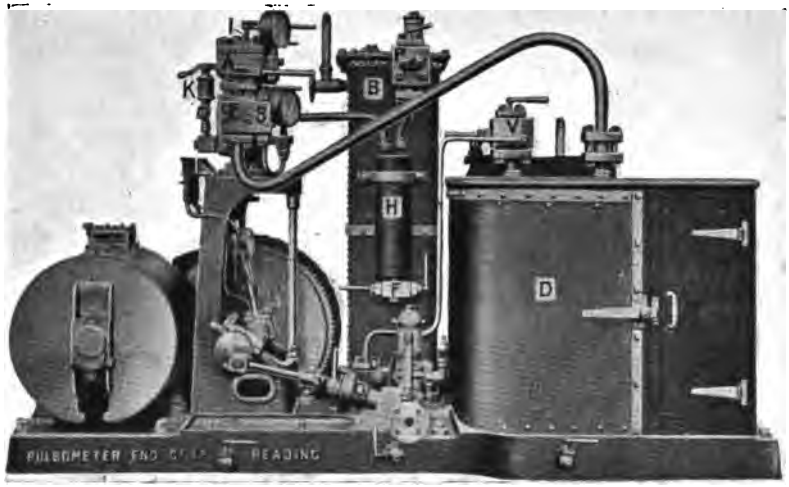


FIG. 319.—Ammonia Compression Refrigerator.

is allowed to accumulate inside the condensing coil until a slight head of liquid pressure is produced. Under this small head the liquid is allowed to leak through a small pipe from the header *L* to the valve *M*, which admits it into a coil immersed in brine. The increase in volume on entry to the brine cooling coils allows the liquid to expand and gasify at a temperature of about 10° Fahr., while the heat required for the conversion of the liquid into a gas is drawn from the surrounding coils and brine. The quantity of heat thus extracted from the brine is nearly equal to the quantity given out by the compressed gas to the condensing water during liquefaction, and the difference is equal to the work done by the steam engine in the same time. The gas returns to the compressor through the pipe *N*, and, being again compressed, commences a fresh cycle of operations.

[FIG. 319A



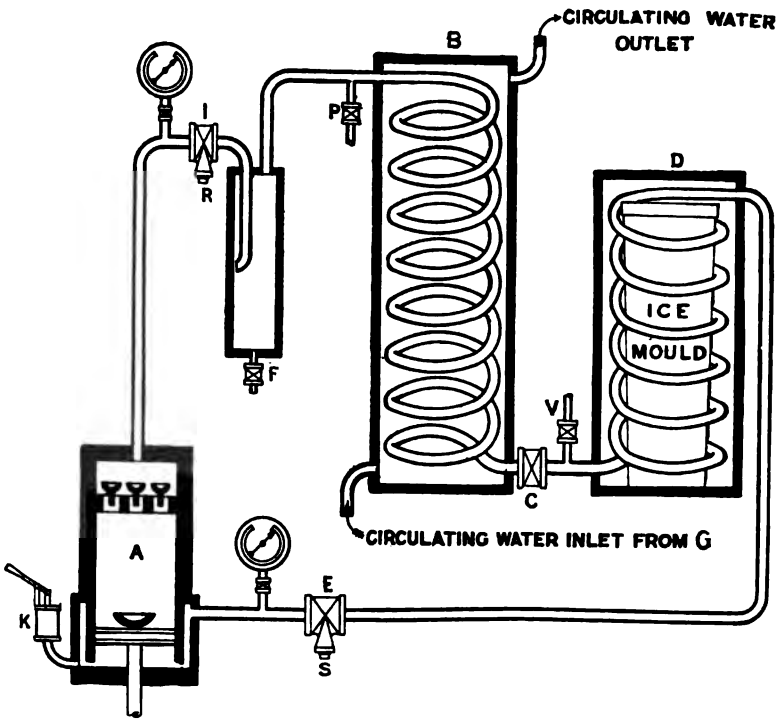


FIG. 319A.—Ammonia Compression Ice-making Machine.

The cooled brine is circulated by a pump through a series of pipes, arranged above and around the refrigerating chamber, and is then returned to the brine cooler for another cycle of operations. Air is sometimes used instead of brine, and is then circulated through a circuit of air trunks by a fan and returned to the cooling chamber. It is necessary to use brine, instead of water, to resist freezing; and salt is added to pure water until one gallon of brine weighs about 13 lb. (water equals 10 lb.).

No copper, or copper alloy, can be used in contact with ammonia.

Fig. 319A shows an elevation and diagram of an electrically-driven anhydrous ammonia (NH_3) compression ice-making machine made by the Pulsometer Engineering Company. Many of these machines are also steam-driven, but electrical driving is frequently more convenient. The following particulars refer to the figure :—

A, Compressor.
B, Condensing tank and coil.
C, Regulating and expansion valve.
D, Refrigerating tank, coil, and ice-mould.
E, Suction stop valve (non-return).
F, Blow-off valve on *H*.
G, Circulating water pump.
H, Receiver or oil separator.
I, Discharge stop valve.
K, Oil pump.
P, Foul gas valve.
R, *S*, Drain and clearing plugs.
V, Charging valve.

The annexed table shows the equivalent boiling-points and absolute pressures of NH_3 (liquid).

The *suction* pressure, shown by the gauge near *E*, should be such that the boiling-point should be 10° F. *below* the temperature of the *brine* contained in *D*.

The discharge pressure, shown by the gauge near *I*, should be such that the boiling-point should be 10° F. *above* the temperature of the *sea* (circulating) inlet *water* at *G*.

Boiling-Point.	Absolute Pressure.	Boiling-Point.	Absolute Pressure.
Zero F.	30 lb. per sq. in.	40°	73 lb. per sq. in.
+ 5° F.	34 „ „	50°	89 „ „
10°	38 „ „	60°	107 „ „
15°	42 „ „	70°	130 „ „
20°	48 „ „	80°	154 „ „
25°	53 „ „	90°	182 „ „
30°	60 „ „		

Carbonic Anhydride System.—Carbonic anhydride, CO_2, under normal atmospheric pressure and temperature, exists as a gas, but if compressed to about 34 atmospheres (or about 500 lb. per square inch), it becomes liquid at a reduced temperature of about 30° Fahr. In other words, the boiling-point of carbonic anhydride, under a pressure of 34 atmospheres, is about 30° Fahr., while under a pressure of 1 atmosphere it is about −120° Fahr.

The general system of refrigeration is similar to that used for ammonia, and is shown in Fig. 320, which does not, however, represent any particular plant. The gas is first compressed to a pressure of about 55 atmospheres (825 lb. per square inch),

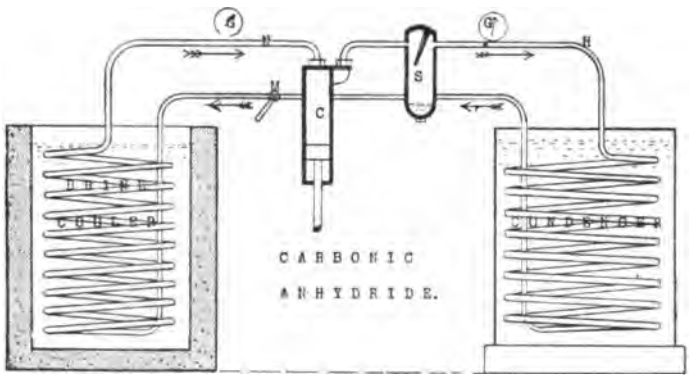


FIG. 320.—CO_2 Refrigerator.

according to the temperature of the surroundings and of the gas-supply. The compressed gas passes from the compressor *C*, through an oil separator, *S*, and a pipe, *H*, into the condenser coil, where it liquefies. The liquid is allowed to leak through the valve *M*, and, expanding into the brine-cooling coils, is converted into a gas while extracting from the brine the necessary heat for its conversion. The gas returns to the compressor at a pressure of about 25 atmospheres (375 lb. per square inch). If the available temperature of the condensing water is high, the pressure must be correspondingly high throughout the system; this is also common to all systems.

The pressures used with the carbonic anhydride refrigerator are necessarily very great, and some difficulty is generally experienced in keeping the packing glands and joints tight enough to prevent leakage. The fumes of CO_2 are, however, not so offensive

as those from ammonia, and the machine can be fitted in the engine room without any extraordinary risk. The system is not so economical as the ammonia type, as the work done in compressing the gas is greater, but the space occupied is less.

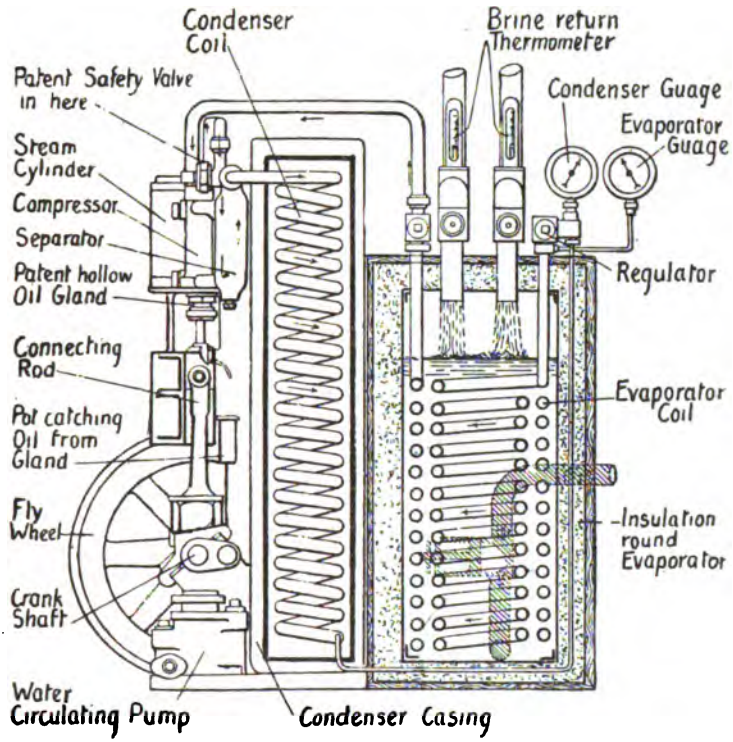


FIG. 321.—Hall CO_2 Refrigerator.

Fig. 321 shows a sectional elevation through a Hall vertical marine CO_2 refrigerating machine. Many refrigerators are now driven electrically.

The condenser coils are made of copper, the compressor of bronze, and the brine-cooling coils are made of iron and fitted in a wrought-iron casing.

[FIG. 321A

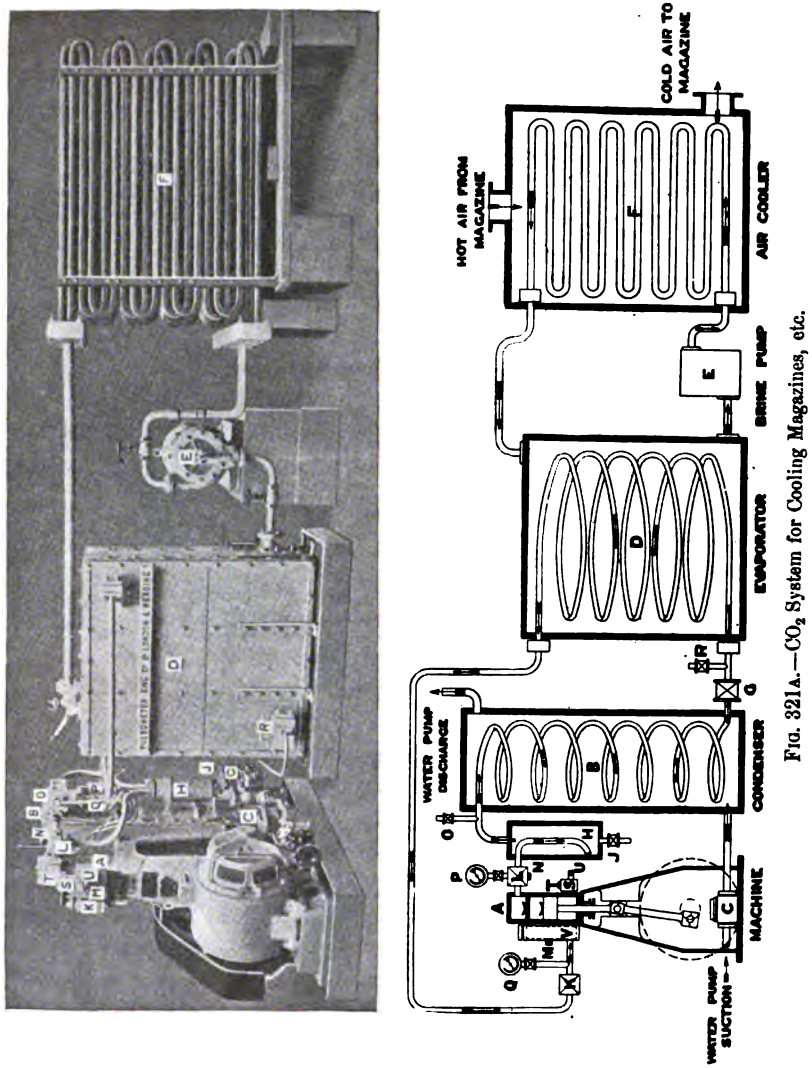


FIG. 321A.—CO_2 System for Cooling Magazines, etc.

Fig. 321A shows an elevation and a diagram of a CO_2 refrigerating plant constructed by the Pulsometer Engineering Company for cooling magazines, etc. The apparatus may be driven either electrically, as shown, or by steam or other power. The general principle has already been described, and the parts can be followed from the following:—

REFERENCE TO FIG. 321A

A, The compressor which compresses the CO_2 *gas*.
B, The condenser, in which, in conjunction with the compression already obtained, the gas is converted to a liquid by cooling.
C, The circulating pump used for cooling the condenser coil.
D, The evaporator in which the liquid CO_2 is evaporated into a gas (at about 28 atmospheres' pressure). The heat required for the conversion of the liquid into a gas is supplied by the *brine*, which is cooled in consequence to a lower temperature. (When in full work the temperature of brine on entry is 40° F.; and on leaving, 38° F.)
E, The brine pump for circulating the brine (weight = 11 lb. per gallon) continuously through the evaporator *D* and air-cooling coils *F*.
F, Air-cooling coils.
G, Regulating valve.
H, Receiver and oil separator.
J, Oil drain from separator.
K, *L*, Stop valves.
M, *N*, Plugs for clearing purposes.
O, Foul gas outlet.
P, Condenser pressure gauge.
Q, Evaporator pressure gauge.
R, Charging valve.
S, Lubricator.
T, Filling plug for *S*.
U, Lubricator valve.
V, Gauge strainer in suction block.

Full instructions are issued with the apparatus, and these should be carefully followed until experience is gained in the actual working.

The Compressed-air System.—This system was generally used for warships, although less economical than either the ammonia or the carbonic systems. When these latter substances are used as the cooling medium, a supply must be carried in the ship, generally in sealed cylinders under compression. Warships are frequently for long periods away from a base of supply; and the compressed-air system has been fitted in several ships, but the CO_2 system is now usually adopted.

The air is not liquefied, as the gas is in the other systems, and

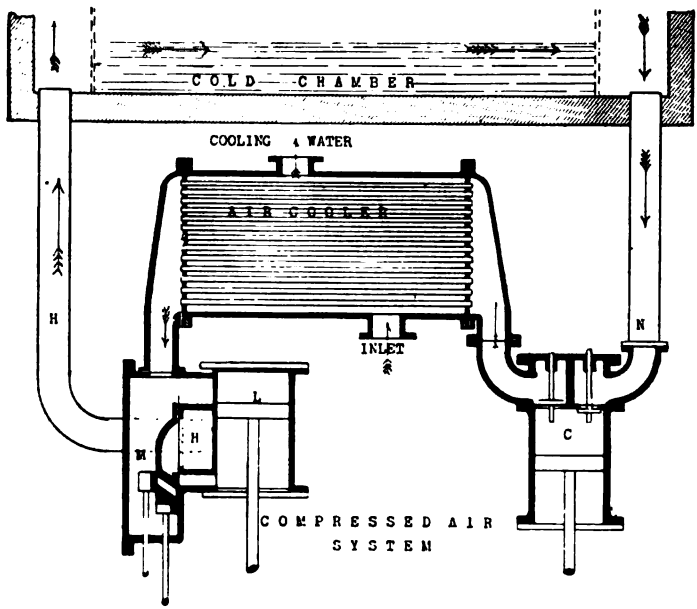


FIG. 322.—Air Compression Refrigerator.

the reduction of temperature is obtained by conversion of heat into mechanical work. The air is first compressed to about 10 atmospheres, and the compression increases its temperature (as with all gases), and then passes into an air-cooler. The compression cylinder *C* is sometimes fitted with a water jacket. The air-cooler is very similar to a surface condenser; the air passes through the tubes and the cooling water around them. From the cooler, as shown in Fig. 322, the compressed air passes into the slide chest *M* of an expansion cylinder, *L*, where it assists the steam engine in doing the work of compression. A large quantity of the heat

remaining in the air is converted into work, and the temperature is therefore reduced considerably, generally to about -80° Fahr. At this temperature and at about atmospheric pressure, it enters the air trunk, at one side near the top of the refrigerating chamber; the air escapes through holes cut in the trunk, and, being cold, is heavy, and sinks.[1] As it circulates downwards and over the parts cooled, the temperature rises and promotes an upward circulation into a similarly constructed return trunk on the opposite side of the chamber. From the return trunk the air is again drawn into the compressor at about a temperature of 20° Fahr., and a new cycle of operations begins.

Drying pipes are sometimes fitted between the cooler and expander; the air passes through pipes which are surrounded by the return air, the compressed air is cooled, and any moisture deposited. Any slight accumulation of water can be discharged through a valve fitted for the purpose. The air trunks for a large installation are about 2 feet square.

There are many points in connection with the practical working of refrigerators which are of great importance, and these are generally noted in the instructions issued by the makers. In each system the success of refrigeration depends on certain simple laws of conversion and work If the temperature of the incoming gas or air is high, a correspondingly high pressure must be employed, and, in the case of air, a very great ratio of expansion. If the circulating water supplied to the cooler is warm, as in the Tropics, a correspondingly high pressure must be obtained in the compressor. Therefore, in practice, success is fairly certain if the necessary high pressures are obtainable with the plant.

In the compressed-air system the temperature of the air supplied to the compressor is of the first importance, and if, as in some cases, the air is taken from the surrounding atmosphere in a very warm compartment, means must be devised to reduce the temperature of the compartment. Generally some circuit of trunks can be built so that the air travels in a continuous stream through successive cycles of operations, and by this means a low temperature can be obtained after working for a moderate time. In ordinary mercantile practice it is generally necessary

[1] 1 cubic foot of dry air at 32° F. weighs 0·08073 lb.
1 ,, ,, ,, 20° F. ,, 0·08274 ,,
1 ,, ,, ,, -80° F. ,, 0·10446 ,,
Thus, air at -80° F. weighs 1¼ times that at 20° F.

to stop the engine about every four to eight hours to clear the ice from the slide chest of the expanding cylinder, and to allow some portion of the ice, which is not detachable, to thaw. The slide-chest cover is generally made for quick and easy removal.

Notes on Refrigeration.—The thermal efficiency of a heat engine rises with the difference of the temperature of the working substance supplied and that at which it ceases to do work in the engine. Conversely, the less the rise in temperature of a gas during compression, as in refrigerating machines, the greater is the efficiency of compression, because the engine or motor has less work to do in compressing the gas.

Generally the mechanical and overall efficiencies of refrigerating machines are fairly high, and the work done by the engine is a rough, *but excessive*, measure of the heat extracted from the brine. Thermodynamically the three vapour-compression machines described above should have about the same efficiency, but practically there is a considerable difference caused by variation in mechanical arrangement. Many of the steam-driven plants use as much as 50 or 60 lb. weight of steam per I.H.P. per hour; one H.P. is represented by 42·3 b.t. units, and to convert 1 lb. of water at a temperature of 32° F. into ice at the same temperature 144 b.t. units are at least required. Therefore, a 20-H.P. engine under these conditions would produce less than 6 lb. of ice per hour, and in its production nearly 1 cwt. of coal would be burnt. For some cold-air machines this is by no means a high estimate; but, happily, improved methods and a better knowledge of working conditions now enable 1 lb. of ice to be produced from water at ordinary temperatures for about 5 lb. of coal with either an ammonia or a CO_2 system.

CHAPTER XXX

ELECTRICAL AND OTHER AUXILIARY MACHINERY

Recapitulation.—The auxiliary engines necessary to the main propelling engines and boilers are :—

1. *Reversing engine*, generally of the two-cylinder type, with cranks at right angles, for moving the links to and fro, as required, into the ahead, astern, or linked-up positions, and is controlled by a differential valve. For very small plants, hand-reversing gear only is fitted ; but for larger engines, hand gear is always fitted in addition to steam gear. In the mercantile marine a single cylinder engine is usually fitted in conjunction with Brown's gear (*see* Fig. 186). In a few modern warships the reversing engine can also be coupled up with the turning gear and used to turn engines in harbour.

2. *Circulating engines*, for driving the pumps generally of the centrifugal type, used for circulating the cooling water through the condensers. The pumps can also be used for removing water from the engine and boiler rooms.

3. *Air pumping engines*, now of great power and generally worked by independent engines, for creating a vacuum in the condensers in conjunction with the cooling action caused by the circulating pumps.

4. *Feed pumping engines*, now generally independent of the main engines and usually situated in the boiler rooms. Originally the main feed pumps were worked by rocking levers from the main engines, and the auxiliaries only were worked independently.

5. *Steering engine*, which is necessary for manœuvring the ship. Hand gear is also generally fitted for use in case of derangement of the steering control or engine.

6. *Evaporator feed and brine pumping engines*, fitted in connection with each evaporator.

7. *Distiller circulating and fresh-water pumping engine,* fitted in connection with each distiller.

8. *Turning engine,* for use in harbour to move the main engines and for emergency repair at sea. An electrically driven motor is now commonly fitted; and when steam is down, electric power can generally be obtained from a dockyard, which makes the system a very convenient one.

The next most important are :—

9. *Capstan engine,* for raising, and, if necessary, catting the anchor; for warping the ship, and occasionally for hoisting in cargo and ammunition.

10. *Fire and Bilge Pumping Engines.*—Originally, like the air and main feed pumps, the fire and bilge pumps were worked by rocking levers from the main engines, but are now self-contained.

11. *Latrine Pumping Engine.*—A small engine, of the same type as the fire engine, is commonly fitted in the larger ships for pumping through the fire-main and flushing the latrines.

12. *Ventilating engines,* used for driving fans for general ventilation and to supply air for combustion. The fans (for forced draught, as well as for purely ventilating purposes) are now generally electrically driven; but the motors, when situated in the upper parts of the boiler rooms, suffer considerably from the high temperature in those parts, and are consequently not entirely satisfactory unless proper arrangement has been made for keeping them cool at all times.

13. *Workshop engine,* for driving the machinery in the workshop generally fitted in warships. The engine generally rotates the driving shaft in one direction only, and has two cylinders with cranks at right angles. Electric motors are now usually fitted, either to drive the shafting in place of the engine, or a separate motor may be fitted for any special machine, as in many shore workshops. If a steam engine is fitted, the heat from it makes the workshop compartment uncomfortably hot, and therefore electric driving is preferable.

14. *Ash-hoisting Engines.*—These are generally of the two-cylinder type, with a differential valve. Automatic stopping gear and a brake are fitted, so that the hoist can be regulated. Electric motors are now sometimes fitted for this purpose, with worm gearing to make them self-holding. (See also Ash Ejector, Chapter IX.)

15. *Refrigerating engines,* used in connection with various

refrigerating machines. In many recent warships these machines are electrically driven.

16. *Power engines*, for generating electric, hydraulic, or compressed-air power for transmission to secondary motors, and for electric lighting purposes.

If all the auxiliary engines and hydraulic, electrical, and air motors in a modern battleship were worked at full power *at the same time*, they would develop about 15,000 horse-power. The total number of steam engines is about 100, of which 20 are used to supply power to about 275 secondary motors. The power engines and many of the motors are fitted in duplicate in case of a breakdown. It is seldom that even one complete set is in use at the same time, and, consequently, under normal fighting conditions only two boilers, which are each capable of supplying about 1000 I.H.P. to the propelling engines, usually provide the power required for auxiliary purposes.

Secondary Motors and Efficiency of Transmission.—Electric energy is not yet obtainable in large quantity from a natural source, and consequently it is necessary to employ a *prime mover*, either the steam or gas or oil engine on shipboard (or natural hydraulic power from a waterfall on land), to generate it by driving a dynamo. The average efficiency, or ratio of Brake horse-power of the shaft to the I.H.P., of a steam *engine* used for driving a dynamo is about 66 per cent; in a few cases it may be as great as 80, but it is more frequently less than 65 per cent. Only about 90 to 95 per cent of the energy of rotation of the engine shaft is converted into electrical energy at the dynamo terminals. Further reductions take place in transmitting electrical energy at the terminals to a motor, and only about 90 per cent of such energy is transmitted to the motor shaft. The electric motor being a high speed machine, it is frequently necessary to employ speed reduction gearing, and consequently there is a further loss of from 30 to 70 per cent (in a bad case).

The ratio of the useful work obtained, by first generating electricity and then applying it to a certain purpose, to the energy expended in producing it, is therefore—

= efficiency of *engine* × efficiency of dynamo × efficiency of motor × efficiency of gearing

= 66 per cent × 90 per cent × 90 per cent × 70 to 30 per cent

= $37\frac{1}{2}$ per cent to 16 per cent.

This result compares very indifferently with the efficiency of the steam engine directly applied, and with the combined efficiency of the engine and gearing employed. If a screw-worm and spur-wheel gearing be employed (a very general self-holding gear), the efficiency of such gearing is not usually above 45 per cent; and as the motor runs at a great rate of revolution, the efficiency is probably less. Thus, for comparison—

Efficiency of engine applied directly = 66 per cent;
Efficiency of engine and gearing = $\frac{66}{100} \times \frac{45}{100} = 31$ per cent.

That is, the efficiency of a steam engine, a prime mover, is generally nearly twice as great as the total efficiency obtained by using a *derived* or secondary motor power, such as electricity. These figures are borne out in everyday practice.

The use of secondary power may be considered under three headings:—

1. *Convenience.*—There are certain parts of the ship, outside the machinery and boiler spaces, where a steam engine is inconvenient. The long lead of steam and exhaust pipes causes delay and loss of economy from condensation; and the heat given out by imperfectly lagged pipes is objectionable in certain places, such as ammunition passages and magazines. Neither for working guns in turrets nor in casemates is steam convenient, nor for fitting in these places, and generally electric or hydraulic power is preferable.

2. *Economy of Fuel.*—The steam engines used for ventilating fans and other purposes, *where a rotative motion is imparted,* may be worked at about the same fuel economy by electric motors, if the steam engine used to obtain the derived electric power is sufficiently economical. To make any appreciable gain, the larger engine used for generating purposes must have not more than half the coal consumption per I.H.P. of a steam engine directly applied. These smaller engines consume about $4\frac{1}{2}$ to 6 lb. of coal per I.H.P. per hour, and therefore the coal consumption of the dynamo engine should not be greater than $2\frac{1}{4}$ to 3 lb. This very moderate consumption is only obtainable with the very best types of three-stage expansion engines used for generating electricity. The economic gain is therefore doubtful, and in any case must be very small; there can be no gain if worm and spur-wheel gearing is employed, but a gain is just possible when cogged wheels are used to reduce the rate of rotation.

All pumps, such as feed pumps, fire and bilge pumps, evaporator feed and brine pumps, distiller pumps, air pumps, and air-compressing pumps, have a necessarily slow reciprocating motion, and for the purposes employed are very efficient, because no gearing is used, and the power is directly applied from the steam piston. If driven electrically, the rapidly rotating motion of the motor must be converted by gearing into a comparatively slow reciprocating, to-and-fro motion. There is a consequent drop in overall efficiency of about 50 per cent, and in the case of feed and air-compressing pumps the drop may be as great as 70 per cent. As an example, to keep the pumps working, the feed engines require about 1 per cent of the total power indicated by the main engines, and where Belleville boilers and feed regulators are used, as much as 2 per cent of the total power may be required. The *Good Hope* class, of 30,000 I.H.P., would require about 450 to 600 I.H.P. to drive the feed pumps. An engine of about 125 I.H.P. is required to drive a dynamo of 600 ampères and 105 volts; and for the feed pumps, even assuming unit efficiency, four such dynamos would be required. If we put the efficiency as high as 50 per cent (almost impossible of attainment), eight such dynamos would be required, and this is the least possible number in practice.

3. *Economy of Weight and Space.*—If the pumps enumerated above are driven by electricity, a large increase in the electric generating plant is necessary, and such increase will probably take up space and weight equal to two or three boilers. The feed pumps remain, but the steam cylinders are replaced by motors and gearings. The motors must be fairly close to their work, in the stokeholds, and if well protected from dirt and dust, are subject to heating from lack of ventilation, and are therefore very liable to breakdown. The mere fact of the conversion of the energy of a steam engine into electrical energy through gearing, and then into mechanical work, is direct evidence of increased weight and space over the application of the energy of the steam engine directly applied.

Conclusions.—For certain purposes, outside the machinery and boiler spaces, electric motors may be more suitable than steam engines directly applied; but not within these spaces, except for ventilating fans which have a high rotational speed. If any economy is made in coal consumption, the dynamo engines must be of either the most economical three-stage expansion, or of the

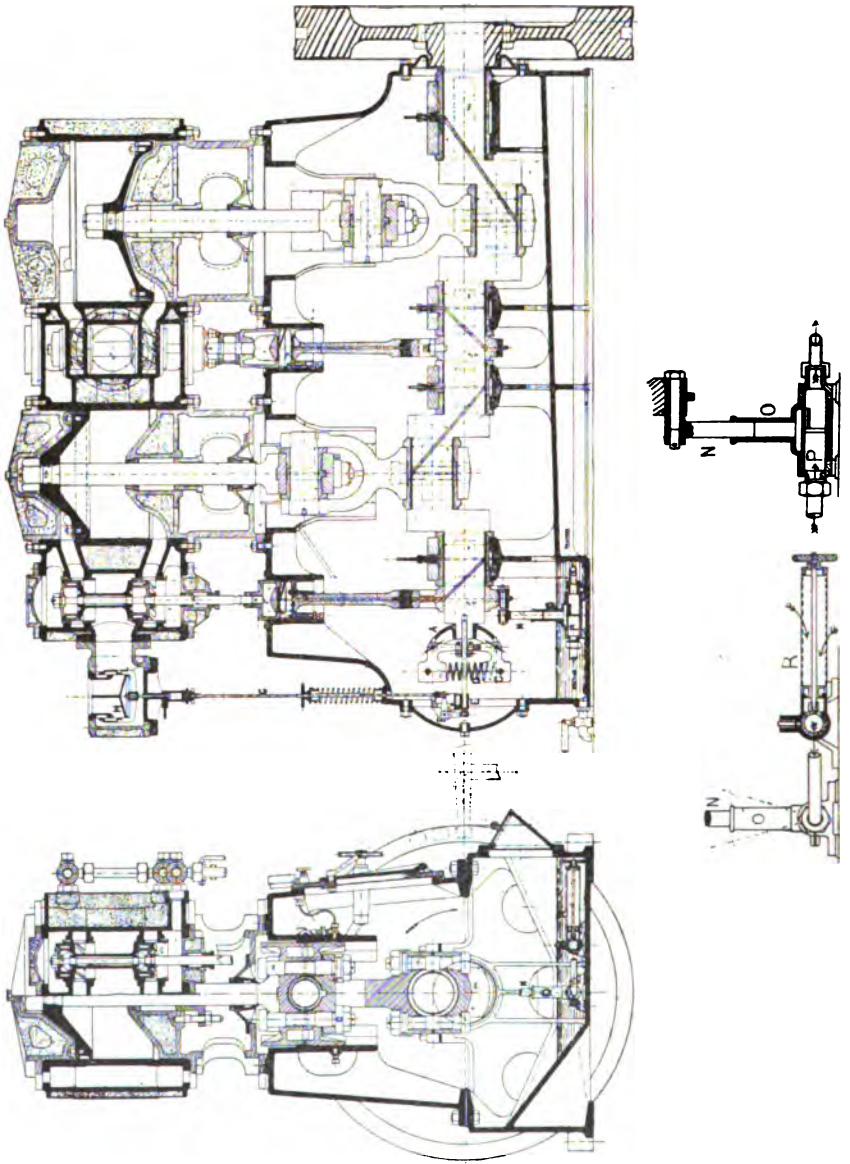


FIG. 323.—Brotherhood Two-stage Expansion Dynamo Engine.

turbine, or of the internal combustion type. The present two-stage expansion engines used for this purpose are not sufficiently economical, and as the power of dynamos is now being largely increased, one of the above-mentioned types should be substituted.

In warships the fuel consumption for auxiliary purposes is very great, and generally amounts to about 10 or 15 per cent of the coal used for propelling purposes at sea. This means a decrease of steaming distance, and consequently only the most efficient *derived* power should be substituted for a *prime mover.*

Dynamo Engine.—In Fig. 323 a two-stage expansion Brotherhood engine, used for driving an electric generator, is shown, and is of about 150 I.H.P. The H.P. slide valve is of the cylindrical type, and two valves of similar construction are used for the L.P. cylinder. All the working parts are proportioned for high-speed running, generally about 400 revolutions per minute.

A *governor* (an enlarged section of which is shown in Fig. 324) is necessary to all dynamo engines, as when working searchlights and motors the load is continually varying, and unless a fairly efficient governor is fitted, the sudden increase of the speed of the engine may produce a sufficient increase of voltage in the dynamo to fuse the armature and other winding wires. In the figure the arms *A* are attached to the crank shaft, and revolve with it. Right-angled levers are pivoted on the ends of the arms, and the ends *B* are in sliding contact with a sleeve piece attached to a small shaft projecting axially from the crank shaft. As the arms move about the pivots the small shaft is moved to and fro. The other ends of the arms *C* are fitted with weights, which are restrained from flying outwards, when the engine is in motion, by a spring, *D*, tending to draw them together.

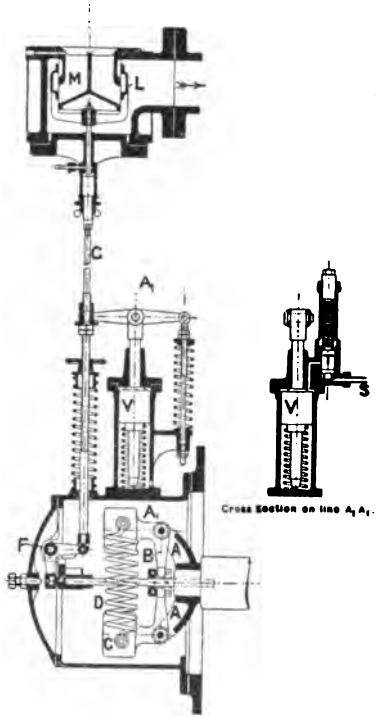


FIG. 324.—Brotherhood Combined Speed and Pressure Governor.

When the speed of rotation of the engine is too great, the centrifugal force of the weights overcomes the spring, and, moving the small shaft to the left, presses on one end of a right-angled lever, pivoted at *F*. The other end of this lever is attached to the rod *G* and valve *L*. When the rod *G* is pulled downwards, it brings the valve with it, and closes the opening on the circular fixed seating *M*. The valve *L* regulates the steam supply to the engine, and is worked entirely by the governor. An ordinary shut-off valve is fitted, above the valve box *M*, in the steam supply pipe. The governor valve rod is of small diameter, and being made a good fit in a long gland, prevents almost entirely all leakage.

For Brotherhood air compressors, the speed governor is of similar construction, but in addition a pressure governor, shown in Fig. 324, is also fitted. *S* is in open communication with the air reservoir, and when the pressure is excessive it lifts the small plunger *T* and admits pressure above the large plunger *V*, which tends to close the valve *L*.

Forced lubrication is fitted to all the working bearings. The pump is valveless (Fig. 323), and consists of a solid plunger *N*, working in an oscillating pump chamber *O*. The suction is immersed in oil, which is strained on its way to the pump, where it enters the cylinder *P*, which is fixed in position. The oscillation of the chamber *O* is such that on the up-stroke of the plunger the oil can enter it, through a slot in the outer part of the fixed cylinder. On the down-stroke the oscillation of the pump chamber closes the suction port and opens another port in connection with a fixed discharge cylinder. From the discharge cylinder the oil is forced, under a pressure of about 40 to 50 lb. per square inch, into the various bearings.

The oil under compression enters a groove cut circumferentially about the middle of the main bearings, through small pipes fitted above the bearings, and is diffused over the whole of the bearing surface. A hole is drilled, as shown, from a main bearing to each crank head, and the oil passes through this hole into a similarly cut groove round the crank-head bearing, and is diffused over the bearing surface. All the bearings are made with oil-containing strips, entirely enveloping the ends of each bearing, but between these strips the usual clearance spaces are left near the lips of each of the top and bottom brasses. The bearing surfaces of the eccentrics are lubricated in a similar way, and a hole through the

centre of the eccentric rod conveys the oil into the top-end bearing. The crosshead brasses are lubricated through a separate pipe, directly connected with the pump delivery. The oil is injected, through holes in the back and front of the guide, into recesses, which are always covered by the guide, made in the slipper opposite the mid-length of the crosshead pin. The recesses are in direct communication with the groove cut around the crosshead brasses. Another connection serves to lubricate each guide, back and front.

The oil pump is single acting, and the supply of oil should be continuous. A relief valve is therefore fitted in the upper part of the delivery pipe, which allows the pressure to escape into a small chamber, which, acting as an accumulator, maintains an almost constant pressure in the various supply pipes led away from it to the bearings.

The oil gradually works its way through the ends of the bearings, falls into the crank pit, and, draining back to the oil-pump suction, is used over again after passing through the strainer. The strainers are of special design, and two are generally fitted, so that one can be removed and cleaned at a time. In the figure, R is the strainer, made of fine wire gauze, fitted over brass end pieces, joined together by a screwed rod, and is removable by unscrewing the rod. When the end of the strainer is lifted it automatically closes the suction opening, and the wire gauze can be removed and cleaned. When the strainer is replaced and turned down into the horizontal position, the suction is automatically opened.

The whole of the lower part of the engine is enclosed by the framing, but doors are fitted for access to all the working parts. Water dripping, or working, through the piston-rod glands is prevented from getting inside this closed chamber by shallow glands, packed with soft packing. The governor end of the chamber is closed by a removable cover for access to the governor. The flywheel end is fitted with a disc, to throw off the oil and prevent it working along the shaft; and a fixed dish-shaped flange is fitted to catch the oil as it is thrown off, and to return it by gravity to the bottom of the chamber.

Forced lubrication is usually fitted to all fast-running auxiliary engines, and is now fitted in many vessels for the main propelling engines, as a result of experiments in a destroyer. Some dynamo engines have been running for some years without appreciable wear of the bearing surfaces.

In the *Belliss and Morcom* two-cylinder *dynamo engine* only one

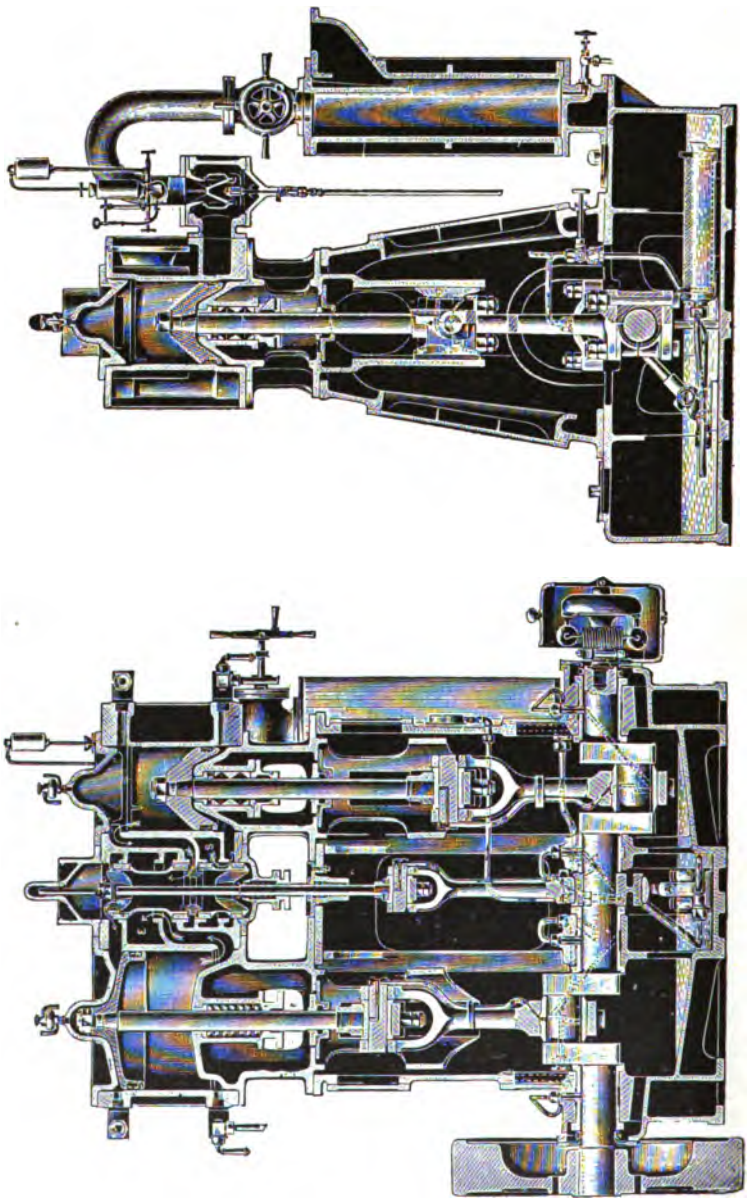


FIG. 325.—Belliss Two-stage Expansion Engine.

eccentric is employed to drive the two slide valves, which are in

one piece, and attached to the same rod as shown in Fig. 325. The disposal of the cranks allows this to be done without any complications. The cranks are always fitted opposite to each other in the two-cylinder engines used for dynamos. This produces less vibration and more uniform running, as the reciprocating weights are more nearly balanced. Messrs. Belliss and Morcom are, I believe, the pioneers of forced lubrication; but the system differs a little from that above described, although the same in general principle.

Some interesting tests of a three-stage expansion dynamo engine (Fig. 326) were recently made by them on the effect of superheat, and they found there was a gain of—

10	per cent	in	economy	by	superheating	50° Fahr.
17½	„	„	„	„	100°	„
23	„	„	„	„	150°	„
27½	„	„	„	„	200°	„

with a corresponding total reduction of the weight of steam used per kilowatt-hour of 6½ lb. (from 23½ to 17 lb.).

Engine Power required for Driving a Dynamo.—The electrical horse-power of a dynamo is equal to the current (in ampères) multiplied by the difference of potential (in volts) and divided by the number 746. The overall efficiency may be taken as about 66 per cent, and therefore—

$$\text{I.H.P. of engine} = \text{ampères} \times \text{volts} \div 746 \times 66 \text{ per cent}$$
$$= \text{ampères} \times \text{volts} \div 500, \text{ about.}$$

EXAMPLE.—Suppose a current of 1000 ampères and a voltage of 120 is required; then—

$$\text{I.H.P. of engine} = 1000 \times 120 \div 500 = 240.$$

The approximate mechanical efficiency of any reciprocating dynamo engine is easily found, because the electrical horse-power is always obtainable from the instruments supplied, and the I.H.P. can be obtained by taking diagrams.

Auxiliary Condenser, Air Pump, and Circulating Pump.—Until recently an auxiliary condensing plant was usually fitted in each engine room of the larger warships, and a separate engine was used to drive the air pump and circulating pump. After the adoption of independent air pumps for the main condensers, two of which are now fitted for each propelling engine, the addition of a separate auxiliary condensing plant became unnecessary.

The construction of the auxiliary condenser is exactly similar

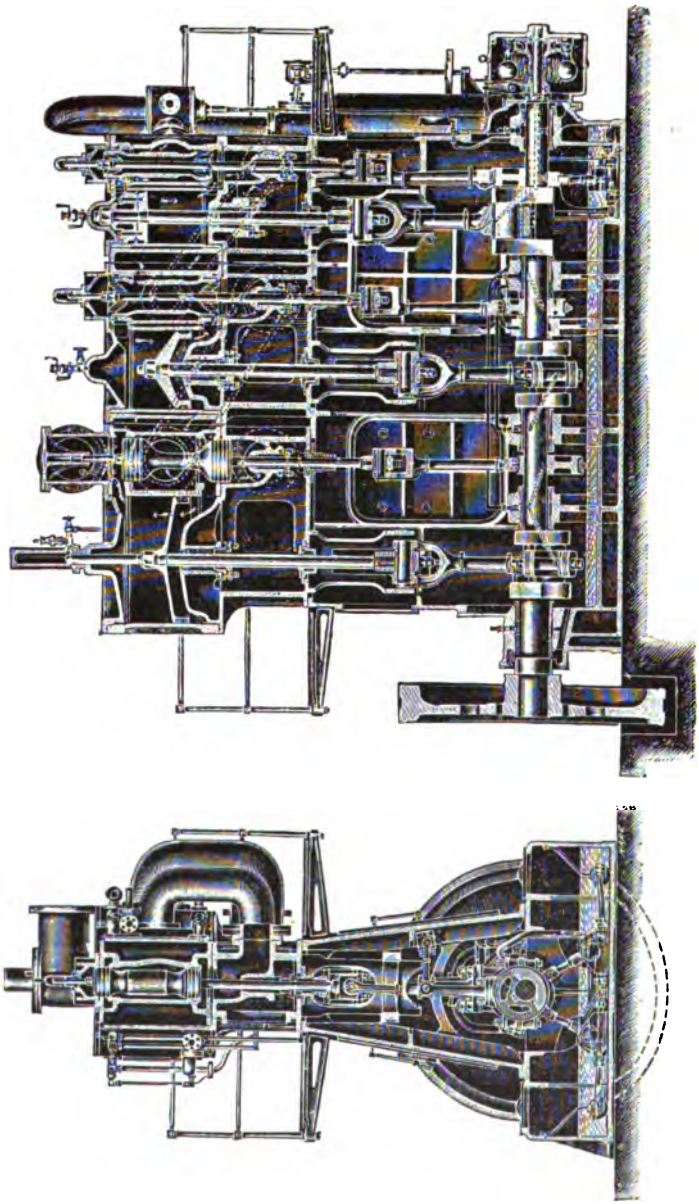


Fig. 326.—Belliss Three-stage Expansion Engine.

to that of the main condenser. The reciprocating air pump and

rotating circulating pump, when driven by the same engine, are not capable of great variation in load without loss in efficiency. This is, perhaps, not of great consequence in itself; but the wear and tear caused by generally overdriving the air pump, in endeavouring to keep the condenser cool by increasing the speed of the circulating pump, is very considerable. The air pumps in the more recently constructed ships are of the solid plunger type, either single or double acting, and the wear and tear is not so marked as when the bucket-valve type is used.

Forced Draught and Ventilating Fans.—The centrifugal type of ventilating fan is used for engine and boiler spaces; but outside these spaces, when driven by electric motors, small helical fans are used. The centrifugal fan is exactly similar in principle to the centrifugal impeller used for circulating water through the condensers; but as air is less dense than water, and very much larger quantities must be passed through the fans, the impeller is much longer (axially), with consequent larger inlet and outlet openings.

When directly steam-driven, the engine is generally of the single-cylinder type, with forced lubrication for the working bearings. In all recent engines the crank shafting and bearings are enclosed, but precautions are taken that the condensation water cannot drip into the oil well, and, reversely, that the oil cannot find its way into the cylinders from the crank pits.

Turbo-Electric Generating Engines.—In several recently constructed vessels turbo-electric generating engines are fitted. The Parsons turbine engine, shown in Fig. 327, is one of this type, and its action will be better understood after studying Chapter XXXIV. In the meantime, the more important accessories will be described. The speed governor, *M* (and gearing illustrated in Figs. 327 and 328), is of the usual centrifugal type, and is rotated at just below one-tenth the rate of the engine shaft by means of a worm wheel *E* and screw *F*, and by bevel gearing (not shown) driven by the shaft *N*. This shaft, by means of a crank, drives a forced lubrication oil pump, and also a belt pulley for a tachometer, as well as a second pair of bevel wheels and a second governor. One governor controls the total supply of steam through the valve *C*, while the other admits this supply intermittently (in gusts) at full pressure through the valve *B*. Both the valves, *B* and *C*, are of the balanced or double-beat type. Above the valve *B*

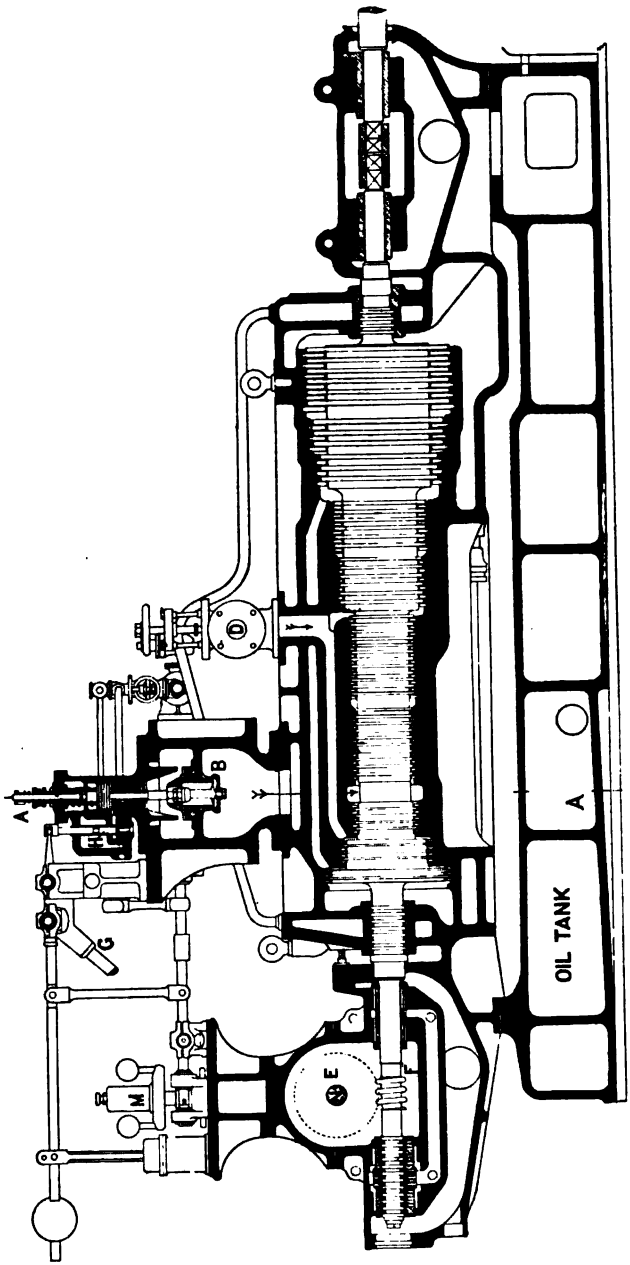


FIG. 327.—Turbo-Electric Generating Engine.

is a cylinder and piston; the piston is pressed downwards by a strong spring, while the steam pressure below tends to force it upwards and thus close the valve. Steam pressure gains access to the space below the piston through a clearance space around the valve-spindle when the valve is opened, but when the valve is closed this clearance passage is also closed. Movement of the rod *G* by the governor-driven gearing produces corresponding movements in the servo-motor (or auxiliary) valve *H*, which, opening or closing the exhaust from below the piston, produces corresponding movements in the valve *B*. Thus greater movement of the governor produces a longer opening of the valve *B*. The valve *B* can be held open, if required, by means of the hand lever shown, and excessive working loads can be carried by the engine by admitting steam directly through *D* to a higher stage of expansion.

The valve *C* is operated by the governor sliding connection or muff and lever acting on the shaft *K* and lever *L*, and the oscillation is steadied by a dash pot and dead weight shown in Fig. 328.

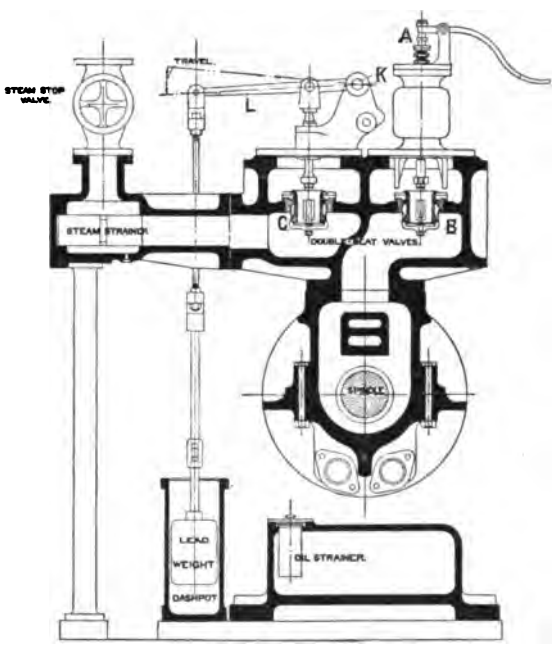


FIG. 328.—Turbine Speed Governor and Regulator.

As will be explained later, a high vacuum is necessary to the economy of turbines, and for this reason a turbo-generator is generally fitted with a separate condenser, circulating pump and air pump. Fig. 329 shows a section through the air pump fitted in connection with the turbine shown in Fig. 327. There are two plungers of different diameters fitted on the same rod. The upper plunger is solid, and the lower one is of the bucket type. The upper plunger ensures the proper closing of the lower pump delivery valves, which, being water-pressure

sealed on the downward stroke, do not leak, and thus prevent a high vacuum being created by the bucket pump.

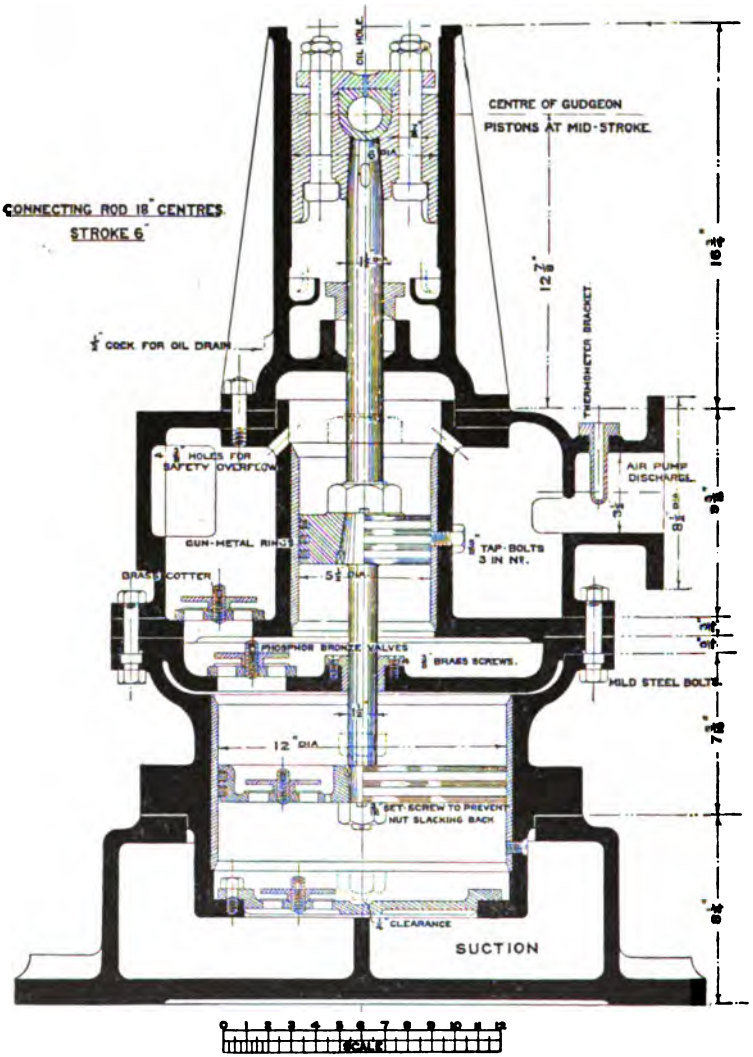


FIG. 329.—Air Pump for Turbo-Electric Generating Engine.

This sketch is fully dimensioned and serves as an example of a fairly good working drawing, which is worth studying and is shown here principally for that purpose.

PART IX

CARE AND MANAGEMENT

CHAPTER XXXI

BOILER PRESERVATION AND REPAIRS

Internal Corrosion.—The principal causes of internal corrosion and decay of boilers are evolution of oxygen, air, and gases from the feed water; keeping the boilers when not in use only partially filled with water; wetness or moisture in a boiler supposed to be empty; dirty condition of the heating surfaces (internally) when under steam; and galvanic action originating between different materials used in their construction.

The general treatment of the internal parts is the same for every type of boiler; but for some boilers the detail is more extensive, and greater care is required because the structural conditions adversely affect the circulation of water and steam. Sea-water contains more air and gas than shore-water. Distilled water contains less than shore-water; and, because it is desirable to exclude air and gases from the feed water as far as practicable, distilled water should be used at all times.

Air, oxygen, and gases are expelled from water by raising its temperature, and therefore hot feed water is preferable to cold; and when either sea or shore water is used for filling the boilers it should be immediately warmed to expel the air and gases, and the air cock or the safety valves should be opened to allow the air to escape before the boiler is shut off for preservation. The boiler should never be allowed to remain open for a longer period than is absolutely necessary for cleaning and repair, and if it is not

immediately convenient to close it up, a wood fire or an airing stove should be lighted in the furnace to keep the general temperature above that of the surrounding atmosphere, and thus keep the boiler dry and free from moisture both internally and externally.

With water-tube boilers, into which cold feed water can be pumped without injuring the boiler from unequal contraction, the boiler is usually filled to the crown and 5 to 10 lb. pressure obtained in it immediately after the fires are out (by using the auxiliary service or the hand pump), and, the water being warmed on entry, the air and gases should be allowed to escape freely before closing up the boiler for preservation. With cylindrical boilers, the addition of a quantity of cold feed water while the boiler is still warm produces racking strains, and therefore the boiler must be allowed to cool slowly; generally it is sufficient to shut the boiler off immediately after the fires are out, and to pump in only a sufficient quantity of water to raise the level to a few inches above the working height, say to a full gauge glass. If it be decided to fill the boiler to the crown, the water should be warmed, and the air, etc. allowed to escape before the boiler is permanently closed for preservation.

Internal corrosion usually takes place in the region of the water-line (working level) or parts of a boiler where the nascent oxygen freed from the water strikes the interior surfaces. The upper parts of the Belleville generator tubes near the junction boxes, the headers of Babcock and other boilers where the circulation is retarded by the change in direction of movement, and the hot water collectors and discharge pipes from Belleville and other economisers, are generally affected in this way.

Corrosive action is checked in Belleville boilers by placing pieces of rolled zinc angle, about 4 feet long, $\frac{3}{16}$-inch thick and $1\frac{1}{4}$ inch $\times$ $1\frac{1}{4}$ inch wide, with the angle uppermost in every economiser tube; and the author recommends that a zinc angle, enclosed in galvanised wire cages, should be placed in each generator *element* in one or two tubes just *above* the working water-level. These precautions, together with others which may recommend themselves in particular cases, and the abandonment of the use of soda, have been found most effective in arresting corrosion.

Galvanic action, originating between different materials used in the construction of the boiler, is checked by suspending *zinc slabs* in electrical contact with various parts of the boiler. The zinc

is attacked by attracting all galvanic action to itself and until it is eaten away, or electrical contact is broken, thus protecting the boiler material. Zinc also counteracts corrosion by corroding away—this is the usual action above the working level,—and it is thus somewhat difficult to determine whether either galvanic action or corrosion causes pitting of the interior surfaces. In any case the careful fitting of zinc slabs acts in a manner beneficial to the life of the boiler material, and additional zinc protectors frequently arrest pitting already observed.

Lime is mixed with the feed water to prevent any acidity—which assists galvanic action—of the water attacking the internal surfaces of the boiler, and as a nucleus for collecting particles of grease. A tank is fitted in the engine room, in connection with the main feed tank, in which a certain proportion of lime—roughly, from 1 lb. of lime for each 1 ton of distilled "make up" feed to 5 lb. of lime for each 1 ton of sea-water "make up" feed—is mixed with a small quantity of water taken from the main feed supply. The clean lime water is allowed to gravitate from the upper part of the lime tank into the feed tank, where it mixes with the feed water and is pumped with it into the boiler. The quantity of lime used should give the water sufficient alkalinity to turn the natural colour of litmus to *blue*. (See Chapter IX., Chemical Tests.) The condensed water, the feed water, and the water in the boilers, are tested for acidity at least once daily, and the lime should be added in small quantities at regular intervals.

A *lime strainer* is fitted on the outlet pipe in connection with the top of each Belleville economiser top (hot water) collector to prevent the passage into the generator feed pipe of particles of dirt or grease collected by the lime. The strainer should be removed and cleaned on each occasion of cleaning the boiler interior.

Soda should never be used in any boiler under ordinary conditions of working. In addition to contributing to the formation of acid (hydrochloric acid particularly), it makes a priming mixture which is very difficult to overcome. Soda in a concentrated form is, however, sometimes mixed with the boiler water and boiled continuously so as to clean the interior of the generating tubes and other parts. The usefulness of this method of cleaning is by no means established, and, with proper filtration of the feed water, should be unnecessary. After using soda in the boiler, it should be carefully washed out with fresh water; or, if a sufficient supply

of fresh water is not available, sea-water may be used for the earlier and rougher part of the operation of washing out.

Periodical Cleaning.—A great *loss of evaporative power* (and fall in economy of fuel) ensues from dirty heating surfaces, and allowance must be made for ready access to periodically clean them. On ordinary service, when clean fresh water only is used for feeding the boilers, and the grease extractors are efficient, every boiler requires a complete internal examination and cleaning at least once in six months; and no boiler should be under steam without examination for more than a total period of thirty days. At such times, every door should be removed and every tube searched and cleaned as far as practicable; the zinc protectors refitted or renewed, and electrical contacts ensured by cleaning and refitting them; and the feed and other valves and the automatic feed regulators cleaned and refitted.

After the internal cleaning of a water-tube boiler is complete, it should be washed through with clean fresh water; then closed up and tested by water to at least the working pressure so as to ensure that all joints are absolutely tight. This test is, of course, supplementary to the periodical water-pressure test, and is usually made after any minor repairs have been effected, such as fitting new nickel ferrules, or making good leaky joints.

When dirty or greasy or brackish feed water is used, the examination and cleaning must be made more frequently; and with cylindrical boilers, particularly those working under forced draught and high pressures and temperatures, the necessity is still more urgent.

The *exterior*, or fire side *of the heating surfaces*, particularly the tubes, require cleaning or sweeping when under steam at least once in twenty-four hours (according to the rate of combustion and class of fuel used).

After steam is down, the external parts of the heating surfaces, etc., require a thorough cleaning with steel wire brushes or other manual appliances, and care should be taken that all dirt, soot, etc., are removed from behind the elements, flame plates, and brickwork, because when under steam the natural expansion of the material of the boiler may otherwise be restricted and thus bring unnecessary stress on the casings and supports; the firing of soot behind these parts also tends to rot and burn away the more permanent parts of the boiler casings. In Belleville boilers the bending of the feed collectors and lower tubes is frequently attributed to accumulations

of dirt, etc., at the back ends of the elements by choking the roller paths.

Lack of facility for manual cleaning of the sooty surfaces of the tubes and heating surfaces of the boiler is one of the principal defects of the design of many well-known water-tube boilers, and in a great degree accounts for the considerable falling off in efficiency and in economy of fuel after fires have been alight only a few hours. Further remarks on this and kindred subjects will be found with the description of the various types of boilers.

It is important that the air casings should be sufficiently tight to prevent the entry of cold air at any point above the lowest part of the heating surface of water-tube boilers; such leakage has been a frequent source of loss of evaporative power, and a cause of smoke formation in ships fitted with Belleville and similar types of boiler.

The rate of transmission of heat in b.t. units per second per one square foot of surface of plating, or heating surface (approximately)

$$= K \times \sqrt{\text{Difference in temperature at opposite faces of plate.}}$$

For steel, $K = 0{\cdot}07$ to $0{\cdot}08$, when the plates are clean, and is, within the limits of ordinary boiler practice, independent of the thickness of the plate.

When the water surface of the plate is dirty and covered with lime, or greasy scale, the difference of temperature between this side of the plate and the water rapidly increases if the rate of evaporation remains constant. With about $\frac{3}{16}$-inch of greasy scale and with high rates of combustion the difference becomes nearly 400° Fahr.

With about $\frac{1}{16}$-inch of clean lime scale, nearly free from grease, the fuel consumption increases by about 15 per cent; the consumption increases almost in direct proportion to the thickness of scale. A loss in the rate of transmission of heat from the presence of scale considerably affects the safe generation of steam at high powers, and has a distinct bearing on the absence of sufficient allowance for unequal expansion.

External Examination.—The principal causes of external defects developing in boilers are: unequal expansion; excessive forcing; inadequate repair; moisture on the surfaces exposed to corrosive action; and galvanic action caused by leakage from electric circuits.

Unequal expansion of cylindrical boilers after the fires are lighted is always provided for, as far as possible, in the design, because the

boiler must undergo some considerable change of form when one part becomes hotter than another. Thus the top of the boiler which is in contact with the steam may have a temperature of about 380° Fahr.; while the bottom, which is exposed to a cold air current and cold water, may be only at 70° to 100° Fahr. It is this difference of temperature, and consequent unequal expansion, which causes practically all the unavoidable defects in boilers. The radius of curvature of the end plates where they are dished to meet the shell plates is made fairly large because the shell tends to expand more than the end plates, being subjected to a greater average temperature. Similarly, as shown in Fig. 26, other parts are dished with large radii of curvature. Attention has already been called to the expansion in length of the furnaces, and the allowance given to the portable and other stays in proximity to them; to the now usual method of fitting bridge, dog, or girder stays in place of vertical stays above the combustion chamber; to the absence of screwed stays below the combustion chambers; and to the flexibility of the furnaces themselves. Too great care cannot be given to these matters, but, unfortunately, they are among those which the draughtsman frequently leaves unconsidered.

An *allowance for unequal expansion* is also necessary to all parts of a water-tube boiler. The generating tubes of the Niclausse and of the Dürr boiler are only rigidly connected at one end, and they are thus free to lengthen; and therefore the expansion of each tube when clean is entirely independent of any other part of the boiler.

In the Belleville boiler, each element is independent of any other and is only securely held at its connections with the steam chest and with the feed collector; the tubes can expand moderately freely in length.

In the Babcock and Wilcox boiler, the allowance for expansion is comparatively little, but in the more recent types there is considerable improvement on those fitted in the Sheldrake. The greater inclination of the tubes to the horizontal compared to the Belleville type, promotes a better circulation with a more moderate temperature, so that on the whole the allowance for expansion may be considered sufficient for rates of combustion up to 35 lb. per square foot of grate, or about 0·8 lb. of coal per one square foot of heating surface.

In boilers of the upright-tube type, the rate of circulation is very rapid and the allowance for expansion is ample if the tubes have a curvature of about $\frac{1}{4}$-inch or over in each foot of length.

Even in a straight-tube Yarrow boiler it is considered that expansion is allowed to some extent by the partial rotation of the water drums on their respective seatings, to which they are not rigidly attached. This does not, however, allow for difference of expansion of tubes in the same row, from front to back, and in all the more recent types of Yarrow boiler fitted in the Navy the tubes in the rows next the fire are made with some curvature.

As steam is raised, all boilers increase in height. The upper part of the upright-tube type is then supported by the tubes from the water chambers. In boilers of the horizontal-tube type, the steam chest sometimes supports the lower parts by suspension, and the boiler framing is then adapted to supporting the steam chest. Chocks and guys are fitted, as with cylindrical boilers, to prevent movement of the heavier parts of the boiler when in a sea-way.

The straight tubes of water-tube boilers *become bent*, to a more or less extent, whether the tube is held either at one end only or at both ends. This bending is caused by unequal expansion of the upper and lower parts of the tube, and may be best illustrated by considering a nearly horizontal tube through which mixed steam and water are circulated when steam is being generated. The upper part of the tube is principally filled with steam and the lower part with water; heat passes comparatively quickly into the water and tends to keep cool the material of the lower part of the tube; but the upper part, although subjected to a lower heating temperature, is generally at a higher temperature than the lower part because steam is a very poor conductor of heat, and therefore does not convey the heat of the upper part of the tube either to itself or the water beneath it. The upper part being thus lengthened more than the lower part, produces hogging or bending of the tube upwards away from the fire (Fig. 330).

Roughly, the average temperature of the metal of a clean tube may be taken as about one-third that of the fire or gases near it; but the rate of circulation and rate of combustion also affect this temperature materially, as well as the cleanliness, or otherwise, of the water surface of the tube. For an ordinary tube next the fire, with steam up in the boiler, the expansion in length of a 7-feet tube would probably be about 0·4 inch; this allows for the expansion of the tube from a cold water temperature, say 100°, up to about 800° Fahr.

Bent *tubes should be straightened* when necessary, and a limited

curvature may be given them towards the fire so as to lessen the continual trouble of straightening them. The curvature should in neither case be allowed to exceed that necessary to keep the *exit* opening above the highest point of the curve, and all points in the

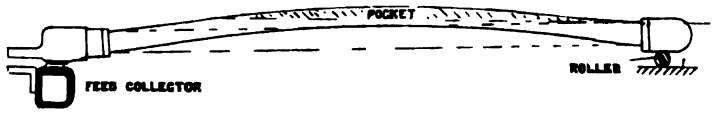


FIG. 330.—Steam Pocket in Bent Belleville Tube.

curve should be above the *inlet* opening; otherwise a steam pocket may be formed (shown shaded in Fig. 330), which tends to produce local overheating of the tube material from insufficient circulation, or from its possible tendency (Belleville type) to reverse the direction of circulation.

Water-tube boilers of the straight tube type require particular observation for deflection after each period of steaming. The curvature is not of much importance in itself unless the tube is left with a tendency to form steam pockets; this applies particularly to the Belleville boiler where the inclination of the tube is only $2\frac{1}{4}^\circ$ to $2\frac{3}{4}^\circ$ from horizontal, and, as a rule, the curvature should not exceed one inch because of the possible formation of a pocket and the tendency to leakage at the nickel ferrule joint.

In the Dürr and Niclausse boilers the generator (external) tubes bend, and it is advisable to straighten them because the internal (circulating) tube remaining straight comes into metallic contact with the generating tube and interferes with the natural circulation.

In the Babcock and Wilcox and the Yarrow boilers, bending or change of curvature is not important so long as it is gradually made in the same direction; but alteration in form tends to fatigue the material, especially when aggravated by alteration of stress. Alteration of curvature occasionally causes leakage at the tube end joints (easily detected when not under steam by presence of scale near the leak), and these should be remedied by lightly expanding the tube into the tube plates. A common defect in Belleville boilers is the bending of the generator feed collector towards the stoking platform. Under ordinary circumstances, free from purely mechanical causes, unequal expansion tends to bend the feed collector in the opposite direction—that is, towards the fire. The weight W of

the element (Fig. 331), which is suspended to the top connection A with the steam drum, exerts a turning moment ($W \times AB$) through the centre of gravity of the element about this point of suspension. This turning moment is almost entirely resisted by the feed collector in a horizontal direction. It is true that the weight of the element may be partly supported by the roller and chair at the back end, but this support is apparently inadequate. When cold, the natural stiffness of the feed collector resists any tendency to bend it; but when heated by the fire it bends under the steady stress produced by the turning moment of the weight of the element about the point of suspension (A, which may coincide with the axis of the steam drum), assisted by any frictional resistance opposed to the natural expansion of the tubes towards the back end.

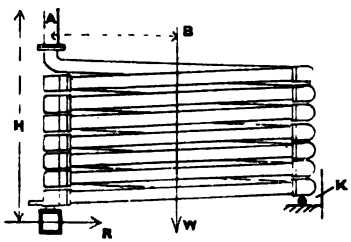


FIG. 331.—Bending Stresses on Feed Collector.

A simple method of protecting the feed collector by brickwork from the intense heat of the furnace is shown in Fig. 332. This method has some advantage over the other methods which have been devised for the same purpose, which generally involve the supporting of the bricks by a bolted attachment—which should always be avoided,—and the angles supporting the bricks are suspended to the lower tubes, which interferes with the examination and renewal of the nickel ferrules. The T.P. method is entirely supported by an angle steel E, resting on brickwork A, and held in place by narrow strips B of flat steel; it does not interfere with any examination or adjustment; the bricks C and D, being of common patterns, can be obtained at any port, and these bricks can be quickly and easily replaced by an ordinary bricklayer at a comparatively slight

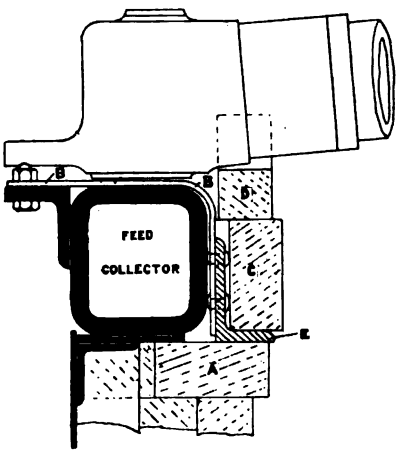


FIG. 332.—T.P. Method of Protecting Feed Collector.

cost. The angle steel *E* may be burnt away, but is more easily and cheaply renewed than the bolted attachments used in other methods.

The *Belleville generator feed collector* tube requires straightening at uncertain intervals of about one year, and the curvature, which is always towards the stoking platform, should not as a rule exceed one inch; otherwise, the back-end junction boxes may be withdrawn from resting on the rollers. The defect may also be caused by accumulation of dirt and ashes at *K* (Fig. 331) between the back casing and junction boxes, which tends to prevent the natural expansion of the tubes when steam is raised. The only remedies are to protect the feed collector by brickwork from the heat of the fire; to light all fires at the same time, thus keeping a uniform temperature within the feed collector; to keep the back-end junction boxes as clear as possible; or to recast the design so that the roller chairs and channel plates become self-clearing or nearly so. To straighten the collector, the usual practice is to remove it by disconnecting it entirely from the boiler. In a few cases they have been straightened in place with fairly satisfactory results, but this can be done more efficiently in a workshop with proper appliances.

The feed collector (Fig. 333) is held squarely to the nipple connections by square sockets in the casings *F* supporting it at each end. It is necessary to reset the sockets occasionally to prevent leakage at the nickel ferrule joints, and the usual fittings allow this to be done fairly easily. Fig. 333 shows one of the best methods devised for this purpose.

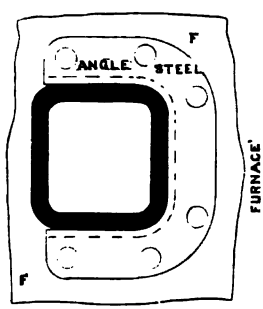


FIG. 333.—Angle Socket Supporting Collector.

Generally speaking, the repair of any water-tube boiler, particularly of the parts subject to steam and water pressure, can be done on ordinary service by the ship's staff, but the repairs to the casings, involving the removal of all elements, are not so easily dealt with. Notwithstanding the brickwork surrounding the furnace, the hot flames and gases, with very slight diminution of temperature, strike the side and back casings and warp and burn them away. This difficulty with casings is common to all boilers of the horizontal tube type, described in Chapter VIII. Although seldom apparent in the early life of boilers, the defect is a constant anxiety and loss of economy after two or three years' ordinary

service. With upright tubes, the casings at the front and back only (which are more easily protected by brickwork) are subject to the hottest flames and gases; but the side casings are subjected to only a moderate temperature, because the gases have been cooled by passing through the passages formed by the generating tubes. This comparison is strongly in favour of the upright tube type.

The essential feature of a *boiler casing* is that it should be air-tight, and the *doors* in connection with it should be properly bedded on asbestos cording so as to maintain the general air-tightness of the casing. Leakage tends to support combustion in the region of the leak, and thus the flames burn away or buckle the casings and increase the tendency to leakage. A supply of cold air, other than that immediately above or below the fire, is not conducive to economy. By proper attention to this and other details, the coal consumption of Belleville boilers in a particular battleship was reduced by over 25 per cent.

In nearly all types of water-tube boilers *baffles* are fitted among the tubes to deflect the flow of the hot gases, and to retard their flow sufficiently to allow them time to communicate their heat to the water in the boiler. It is therefore important that they should be properly placed and in good repair. Baffles are also fitted to check the flow at certain parts of the boiler, between the generating tubes and headers and the casings particularly; these are usually called *flame plates*, and their special use is to prevent the impact of the hot flames on the casing and thus to protect it. They are fitted on all four sides of a Belleville boiler, including particularly the backs and smoke-box doors, and care should be taken that they project sufficiently to fulfil their special purpose. Baffles and flame plates are fitted so that they are easily removable.

The condition of the *brickwork* has an important bearing on the durability of the boiler casings, as well as on the economy of steam generation, and for both reasons it should be kept in the best possible repair. The bricks protecting the back end junction boxes and roller paths of Belleville boilers should be frequently removed so as to clear out all dirt, etc., which may interfere with the free expansion of the element.

Repairs.—So far as the casings are concerned, no special procedure is necessary in making repairs when the defects can be seen and got at. One point should always be borne in mind—that the casings of water-tube boilers, being subject to great changes of

temperature, must necessarily be of a flexible character, and therefore the introduction of stiffening pieces is more likely to increase rather than decrease the seriousness of defects caused in the first instance by buckling from insufficient room for expansion.

In Belleville boilers, *to renew the nickel ferrules* the anchor bolt is slacked off or removed, and, enough space being allowed between the front junction boxes to wedge up the element so that the ferrules can be removed, two new ones are fitted, one above the other. *To renew an element*, it is disconnected at its junction with the top discharge pipe and at the anchor bolt; a roller path of angle steels and a roller are fitted in the furnace towards the back end, below the element, to take its weight; the sight door in the lowest junction box is removed, and a long bar inserted to take the weight at the front end; a piece of wood is inserted between the bar and the upper edge of the hole to prevent damage near the joint; and blocks with differential tackle are attached to the outer end of the bar (where a shackle is fitted) and to some convenient point in the upper part of the stokehold. With a little gentle coaxing the element can be pulled forward and clear of the casing, the weight being alternately supported on slings or chocks as convenient. The smoke-box doors and frames are detachable, and can be removed if in the way. *To renew a tube*, it is generally advisable to remove the element and fit in a spare one until convenient to renew the tube and make the element ready for replacing. If the boiler has been under steam for a few months, it is usually necessary to cut away the check rings and frequently to cut the tube to separate it from the junction boxes, because the screw threads become hard set in place. The renewal of a generator element takes from three to twenty-four hours, according to the time it has been in use. The renewal of an economiser element, being smaller and lighter, may not take quite so long, but the method is similar.

In Babcock and Wilcox boilers, to renew an element the top and bottom nipple connections must be cut through, and a special tool is usually supplied for the purpose. To renew a tube, the element is usually removed and a spare one fitted. The defective tube can then be cut away and a new one fitted at the first convenient opportunity.

In a Niclausse boiler, the process of renewal of an element is similar to that of the Belleville type; but it is seldom necessary,

and more usual to remove the tubes only for examination of the casings, etc.

In a Dürr boiler there are no elements, and tubes can be removed as necessary.

In upright-tube boilers a defective tube can be plugged, as described in Chapter VII.

Durability of Boiler Fittings.—The durability of various particular fittings of Belleville boilers appears to be: for fusible plugs, six months; for nickel ferrules, one year; for brickwork, and flame plating, about one and a half years with minor repairs; and the casings and supports may continue efficient for two years with minor repairs. By taking defects in hand as they appear, and anticipating them (the stitch in time) in a new ship, it is possible to prevent defects accumulating, but it needs continuous attention and intelligent work on a multiplicity of detail which is really a matter of experience and can hardly be mapped out on paper. So much depends on the early treatment, and the work—especially under forced draught—which may be required from the boilers, that any definite durability is extremely difficult to forecast.

At the end of four years it is probable that the joints of all sight holes on the junction boxes of both generators and economisers will require refacing; the channel plates supporting the back ends of the generator elements will require resetting, and probably a small proportion of the rollers and chairs renewing. Dead plates burn away; the ash-pans wear out; the casings open at the joints, leaving air leaks; and the thousand and one details of a Belleville or other water-tube boiler of the horizontal-tube type add to the burden of continual vigilance and repair.

In the upright-tube boiler, the casings can usually be completely examined and repaired without removing any tube or nest of tubes; but in the Belleville type, it is necessary to draw back every element at least once annually to examine the back casings and supports before defects can be ascertained or repairs made. On the other hand, in the upright-tube type the renewal of a few tubes generally entails the removal of part of the casing, whereas in the Belleville and other horizontal-tube boilers any tube or element can be renewed without interfering with the casing. The comparison involves the relative durability of the various parts, and generally the advantage lies with the upright-tube type, because the tubes in any boiler have a longer life than the casings.

CHAPTER XXXII

CARE AND ADJUSTMENT OF MACHINERY

THE main propelling machinery is always erected in the work-shops before its final erection in the ship. By this system the alignment and proper fitting of all parts are ascertained, and any discrepancy remedied while still in the building stage, and usually a water pressure test is made of the cylinders, jackets, and other fittings in the shop. Although this preliminary building up of the engines, only to be taken down and rebuilt, seems somewhat of a useless undertaking, experience shows that a large saving in the cost of labour and better fitting and adjustment are obtained. Before the machinery is erected in the ship it is necessary to bore the stern tube and other bearings, and to make certain that the axis of the shaft, including the crank shaft, lies truly in line. The successful working of the machinery is largely dependent on this accuracy, and very careful sighting arrangements are made for this object alone. This part of the work is usually outside that of the sea-going engineer, and can only be done with sufficient accuracy by using the proper instruments. The sighting is preferably done at night so that there shall be no confused light, and for the same reason it is usual to fit a screen over the apparatus fitted outside the ship. The sighting apparatus consists of a small hole made to exactly coincide with the axis of the foremost bearing, and behind it (at the forward end) an electric lamp of 8 or 16 candle-power is fixed. In each bearing a similar hole, but adjustable both horizontally and vertically, is set, and there is a fixed hole at the aftermost end (stern bracket bearing) which is fitted with an eye-piece. By careful adjustment, using cross-lined sights if necessary, each hole is brought in line between the foremost and after holes until the light shows

clearly from end to end. From these holes the bearing circles can be struck out, and the parts bored accordingly.

Alignment of Shafting.—After the shafting has been fitted in the ship, the alignment can be tested at any convenient time by disconnecting the couplings and drawing the shaft clear from each in turn. The amount of drop, or variation from parallelism of the coupling flanges, can be measured by using feelers, that is, thin sheets of steel graduated for the purpose in one-thousandths of an inch. If the measurement shows any appreciable wear of the bottom brass, it should be removed and re-metalled, so as to bring it up fair with the remainder. A common method of gauging is shown in Fig. 334, and the drop is measured by inserting a feeler between the gauge and the top of the shaft. The diameter of the shaft must also be gauged to obtain an accurate measurement. If remetalling is impossible, the brass may be raised temporarily by means of a liner of the correct thickness placed below it; but this method requires great care and accuracy, and is not generally recommended because any discrepancy in fitting is liable to bring local stress on the brass and tend to distort it. Generally the wear on stern bearings is greater than that at the crank end of the shaft, and consequently there is a droop of the shaft towards the propeller end. If this is general and fairly gradual, no unpleasant consequences need be expected; but if it is excessive, the imperfect alignment brings great alternating stresses on the coupling bolts from the bending of the shaft at every revolution; the frictional resistance is also increased, and there may be a tendency to whipping in very long and small shafts. With large and comparatively short shafts, well supported by bearings at short distances, whipping is not a usual occurrence, and unpleasant experience is seldom occasioned by it.

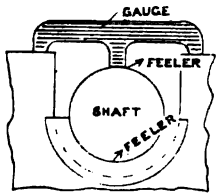


FIG. 334.—Gauging Alignment of Shaft.

Alignment of Pistons, Rods, and Guides.—In *horizontal* engines, the piston rings have a distinct tendency to wear down and allow the body of the piston to come in contact with the liner or working barrel of the cylinder. The alignment can be tested from outside the cylinder by moving the piston to opposite ends of the stroke (*see* Fig. 335 for vertical engine) and carefully measuring the heights A and B of the rod above the guide E in the two

opposite positions. This measurement should be made as near the cylinder as possible, from the same point on the guide surface, and shows any difference in wear of the piston rings and guide. If time permits, it is preferable to open the cylinder and gauge the actual position of the piston with respect to the barrel, and line it up if necessary by adjusting the thickness of the cod piece between the packing ring and body of the piston. When making this adjustment it is usual to set the piston rather above the central position, to allow for subsequent wearing down; this amount should not exceed $\frac{20}{1000}$ths of an inch, or trouble may be experienced with the rod packing from inaccurate alignment of the piston and guide. This alignment should be tested after centering the piston, and if necessary the guide should be lined up also.

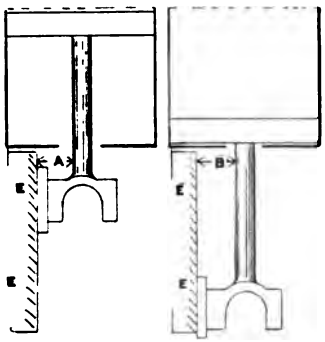


FIG. 335.—Alignment of Piston, Rod. and Guide.

In *vertical engines* the pistons seldom require re-alignment, because the wear is approximately equal all round. The procedure is the same as for horizontal engines, but it is usually the guide that requires attention. With heavy pistons without tail guide rods it is good practice to fit solid blocks in lieu of three or four of the spiral springs in the piston immediately above the guide, and thus prevent oscillation and knocking. These blocks should be carefully fitted so as not to work loose vertically or sideways; mysterious knockings have been stopped in this way which were originally attributed to water hammer or slackness in the guide and crosshead bearing.

The piston packing rings in both horizontal and vertical engines should be allowed sufficient freedom of movement, without leakage, between the flange of the piston and the junk ring. A good practical method of testing this adjustment is to insert, before screwing up the junk ring, slips of thin tissue (cigarette) paper at intervals between the packing and junk rings. When the junk ring is screwed hard up, the slips should be just withdrawable without tearing the paper. If measured by micrometer or soft lead wire, the clearance should not be less than $\frac{2}{1000}$ths of an inch for cast-iron rings of any ordinary width, and of $\frac{1{\cdot}5}{1000}$ths *per inch* of width for gun metal or phosphor bronze.

When the crosshead bearing is disconnected from the piston rod and slipper, it is quite possible for the latter to appear out of alignment with the guide. The slipper always has some clearance sideways, or lateral play, and it is therefore possible for the rod to appear to be out of line. Therefore, this alignment should be tested by applying a set square in the top brass (Fig. 336), and the tongue of the set square should then be parallel to the rod; in making this measurement care should be taken that the rod is parallel, *i.e.* of exactly the same diameter between the points of measurement. If this alignment is apparently correct, *A* and *B* being equal, then connect up the rod and top bearing, and turn engines with the weight of the piston, etc., resting on the top half of the crank bearing. If the alignment is correct, the crank end of the connecting rod will have no movement endways which is not shared by the top end bearing. While testing this alignment, the crank shaft should be prevented from axial movement by nipping one of the thrust-shaft collars fairly tightly between two thrust rings, or by actual observation of any axial movement.

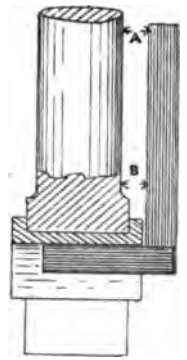


FIG. 336.—Alignment of Connecting-rod Brass.

The *alignment of the connecting rod* is of great importance to efficient working, and should be carefully tested when making adjustments or refitting the connecting-rod bearings. The liners between the top and bottom brasses should be removed from each end in turn, and the brasses tightened up until the rod is just movable by hand; the opposite end is then disconnected, and the rod should swing freely and exactly over its pin. As pointed out above, the guide may be canted sideways (*i.e.* fore and aft), and throw the disconnected end out of alignment; in this case, which is common with heavy engines, the bearings at each end should be tightened up fairly closely (about midway between "hard up" and the usual "working clearance"), and the "marking" (red lead) will clearly show the hard places after giving the shaft a complete revolution. If necessary, the hard places should be eased away from each bearing in turn until the pins bed fairly in the metal at top and bottom for a vertical engine, or front and back for a horizontal engine.

Adjustment of Connecting-rod Bearings.—Although it is not

necessary to test the alignment so frequently as the adjustment, it should be remembered that no adjustment can be entirely satisfactory unless the alignment is good. In each bearing, the pin should bear evenly on the metal of each of the two brasses for a length around the circumference of the pin equal to the diameter of the pin. This leaves $\frac{1}{6}$th of the circumference at each lip of the brass clear of the metal at and near the positions of the liners, as shown in Fig. 337. The thrust and pull exerted through the connecting rod is in the direction of its length, and therefore clearance at the liners reduces frictional resistance without decreasing the effective bearing surface.

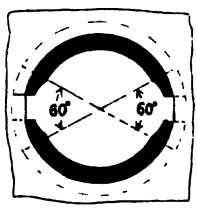


FIG. 337.—Clearance in Bearings.

The adjustment is tested by using *soft lead* wire. Four pieces of wire are generally used for large engines; one piece is laid half round each end of the pin just clear of the oil-containing strips, and one piece fore and aft about one or two inches away from each side of the oilway in the crown of the brass. The bearing is next tightened up to the proposed setting, and then slacked back until the wires can be removed. The thicknesses of the wires are carefully measured by a micrometer gauge and then compared with the usual requirements, which in the Navy are usually recorded in a "Lead book" as below. (Specimen.)

S.M.P. CRANK BEARING

Date.		Marks on Nuts.			Thickness of Leads.		
		Found at	Hard up	Left at		Crown	Horns
8/8/07	Port nut	6 on $1\frac{1}{2}$	6 on $3\frac{3}{4}$	6 on 1	Forward	25	40
	Starboard nut	6 on $1\frac{3}{4}$	6 on 4	6 on $1\frac{1}{4}$	Aft	25	40

Remarks.—Bearing refitted and eased away at horns.
Initials of officer superintending adjustment—K. C. B.

The liners should be of the correct thickness, so that when the bearing is tightened up to the correct adjustment they should be firmly nipped between the butts of the pair of brasses. Before running at full power, the bearings of large engines are usually slacked so as to give about 2 to 3 thousandths of an inch greater clearance. After the bearings are adjusted and tightened up to the required setting, the side play should be tested by a pinch bar,

which can be inserted beside the bearing, and if the adjustment is good the connecting rod should be just capable of easy movement along the pin.

Adjustment of Main Bearings.—This is made in the same way as for connecting-rod bearings, and the soft lead wire can be used and clearance measured in the same way for the top brass. The bottom brass can, if necessary, be removed and bedded on the journal or pin, and the clearance on the horns tested. Care should be taken not to remove metal from the bottom crown of a main bearing brass, because it may thus throw the shaft out of alignment.

Clearance in Bearings.—The thickness of the lead wire taken from the crown of the various bearings should be about—

$\frac{1}{1000}$	inch	per	inch	of diameter of crank pin.
$\frac{1{\cdot}5}{1000}$	"	"	"	crosshead pin.
$\frac{1}{1000}$	"	"	"	main bearing.
$\frac{1{\cdot}2}{1000}$	"	"	"	eccentric sheaves.

The *minimum* clearance for any bearing is about $\frac{1}{100}$th inch, which is about twice the thickness of the film of good oil under pressure.

As a general rule the clearance at the horns should be one and a half times the clearance at the crown, the increase being gradual towards the liners; but the oil-containing strips should have as little clearance as possible. Plummer blocks should also allow some slight clearance at the horns; but the allowance is not so necessary as in connecting-rod bearings, which will not run satisfactorily without a comparatively large amount. When a brass gets warm, the tendency is for it to close at the horns on the pin (as shown in Fig. 338, exaggerated), and consequently nip it closely: which in its turn tends to cause excessive friction, then greater tendency to heating, and next a tendency to close in and nip the pin more tightly. Neglecting to make proper clearance at the horns is a frequent source of overheated bearings.

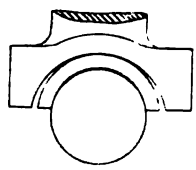


FIG. 338.—Closing of Hot Brasses on the Pin.

Clearance in Piston Packing Rings.—The clearance at the tongue-piece is best determined by calculation of the difference in linear expansion of the circumferential length of the ring and of the cylinder or working barrel. The linear co-efficients of expansion

of the various metals are given in Chapter III., and the different temperatures of the steam can be obtained from the table at the end of Chapter I. The highest and lowest steam pressures must be allowed for in considering the temperature and linear expansion in any particular cylinder. In addition to the above allowance a standard clearance is allowed, based on experience, of about two-thousandths of an inch for each 1 inch of diameter of the cylinder for the same class of material of barrel and ring; that is, say, a cast-iron ring working in a cast-iron barrel or cylinder. Similar clearances must be allowed in cylindrical slide-valve packing rings.

For *piston-rod packing rings* allowance must also be made for wear, and the butts of the segments of metallic packing of the white metal type should have a total clearance of at least $\frac{25}{1000}$ths of an inch for each inch of diameter of the rod. This amount also applies to slide-rod packings.

Thrust Bearings.—The bearing surfaces of the shaft collars should be kept as smooth as possible, and sea-water should be excluded at all times. The clearance between each collar and ring should be carefully gauged, with a properly graduated feeler, to make certain that each ring is taking its fair share of the thrust pressure. Any difference of temperature of the rings when under way points to faulty adjustment. The "ahead" (or after) side of each ring wears away gradually, and thus allows the shaft to move bodily forward; a mark should be made on some easily accessible part of the shaft, and a gauge made relative to some fixed part of a plummer block or other part, so that the amount of wear can be tested at any time at sea or in harbour. If the wear is excessive, it may be necessary to set the thrust rings aft so as to maintain the axial alignment of the connecting rod central to the mid-length of the crank pins.

Air Pump.—The bucket should be an easy working fit in the barrel, especially when the pump is driven at a rapid rate. The plunger packing should not exert a pressure of more than 1 lb. per square inch of its surface in contact with the barrel, and water grooves should be cut about $\frac{1}{8}$th inch wide at intervals of about 1 inch in the width (axial length) of the packing ring or plunger. The air pump should be kept free from grease, the valves should be frequently cleaned and examined, and the securing arrangements and safety stops carefully inspected before the pump is closed up.

Condenser.—The feed-water filters should be frequently cleaned

so as to keep the condenser cooling surface as free as possible from grease. When either soda or potash is used for boiling out the condenser, the mixture of water and soda, or potash, should be carefully excluded from entering the boilers; the author has tried this method of cleaning, and found it inefficient except when the cooling surfaces can be subsequently scrubbed. The only effective method of cleaning a dirty condenser is by removing the tubes and scrubbing each one separately with a strong alkaline solution.

The protecting plates (mild steel slabs) should be examined, and their electrical connection with the body of the condenser should be renewed at least once in six months. The weed traps require constant and regular attention, especially in shallow waters. To prevent corrosion in harbour, the air should be excluded as far as possible from both steam and water surfaces of the condenser, and after steaming it should be left full of circulating water, because this is preferable to draining the condenser imperfectly and leaving beads of moisture on the tube surfaces. Corrosion or galvanic action seldom (the author has never met a single case in his experience) occurs on the steam side, and is almost entirely confined to the circulating water side of the tubes.

Setting of Slide Valves, etc.—It is occasionally necessary to check the adjustment of the various bearings in connection with the eccentrics and link motion, and therefore the slide valves may require resetting. The original clearances of valves (and pistons) are usually recorded, and as a rule the original clearances should be maintained, unless there is a very excellent reason for altering them. It should not be necessary to open out either cylinders or slide chests to check these clearances and settings. A gauge should be made, if not already supplied, for each, and marks made to correspond on the rods and guides for certain defined positions of the pistons and valves.

Any error in the setting of the various slide valves can generally be detected from the indicator diagram, as detailed in Chapter XXI., with various other defects due to leakage and inefficient working.

Fuel-saving Appliances.—All appliances should be used which can save fuel. Only a certain amount of money can be afforded by the nation, or the company, to which the vessel belongs. If a saving of 10 per cent in fuel can be obtained in ten ships, another one can be kept going. Heat is wasted in many ways—

up the funnel by bad stoking; keeping the condenser too cold, and therefore cooling the feed water below the economical limit of about 110°; hot drain water is led into the condenser, where its heat is wasted, instead of being used to heat the feed water; and frequently the pressure in the steam jackets is not properly adjusted to obtain the greatest economy from the steam. It has been already mentioned that the least pressure in any jacket should be equal to the initial pressure in the cylinder; this pressure can be increased with economy so long as no increase of internal lubrication is necessary.

Bilges.—An engine cannot be kept efficient without cleanliness, although it is possible to have cleanliness without efficiency. The health of every one in the ship depends on the cleanliness of the bilges, apart from its purely mechanical necessity. There should be no difficulty in keeping the bilges clean when in harbour; and when at sea the parts which are inaccessible should be washed through occasionally, but the water should be allowed neither to rise above the level necessary to effect this purpose nor to spread itself over an unnecessary area. The principal difficulty with bilges arises from carelessness in cleaning out dirt and clearing out the cotton waste used for cleaning, which clog the mud drums and suctions of the bilge pumps, and thus prevent the bilge water being kept below a normal level. The mud drums at the ends of the suction pipes should be *above* the bottom of the floor, so that the suctions are not choked by the dirt drawn or washed into them.

The engineer, before taking over the watch, should, among other necessary duties, inspect the various bilge suctions to see that they are clear. A good routine is for the petty officer in charge of each compartment to report hourly the depth of water in, and state of, the bilges to the engine room. Precaution should be taken, both at sea and in harbour, that all ashes and dirt are cleaned away after the plates have been lifted, so that the platform lies fairly on the supporting angle steels, or floors.

Water Service.—All bearings on which water service has been used should be opened out and cleaned as soon as possible after arrival in harbour. Although the engines may be turned daily, when steam is not up, the bearing surfaces in the presence of moisture soon corrode and become rough.

The water service should be shut off at least twenty minutes

before entering harbour, so as to leave only clean oil on the surfaces of the bearings when the engines are finished with. An extra allowance of oil is generally made for this purpose, and, if properly used, saves much work in harbour.

Lubrication.—Breakdowns are generally caused by dirt and corrosion, both of which prevent proper lubrication and contribute to inefficient working of the machinery. Dirt can be kept within moderate limits by constant care, and corrosion is assisted by the presence of dirt. In harbour a lot of unnecessary gritty matter finds its way into bearings and working parts, and is a frequent source of trouble. Care should be taken that such gritty matter (emery cloth, emery powder, and brick dust) is not used near any lubricator or bearing. As a precaution, when getting under way, after a general clean-up has been made and insufficient time is allowed to properly clean all the bearings and working parts, the water service should be turned on to each bearing to wash out the gritty matter and leave the bearing surfaces free for clean-oil lubrication. The object of all lubrication is to impose a fluid between the two rubbing surfaces, and so prevent them coming into actual metallic contact. The best method of cutting the oil grooves in connecting-rod and other bearings is shown in Fig. 339 (a rough perspective view of the top and bottom brasses) in connection with oil-containing strips.

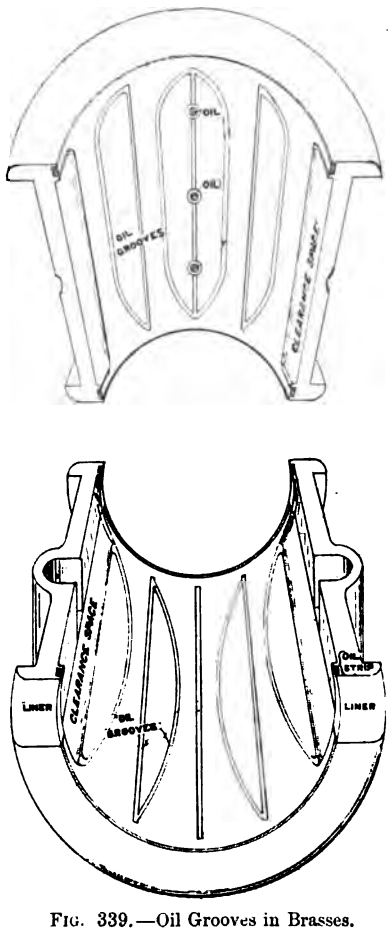


FIG. 339.—Oil Grooves in Brasses.

All lubricators should be scalded out with boiling water and soda, or potash, as late as possible before getting under way, and

afterwards at regular intervals when in harbour. The mixture should not be allowed to find its way into the bearings; and two men are required for the work—one to scald through, and the other to attend the lower end of the lubricator pipe. After this cleansing, clean fresh oil should be poured into the cup and allowed to find its way into the bearing. The worsted and the wire to which it is attached should also be cleaned, or renewed, and made of such a length that the worsted will act efficiently as a syphon.

For external lubrication, the syphon worsted is preferable to any mechanical appliance: it is simple and certain in its action, can be regulated to a nicety, and is very economical. No method, however, compares in either efficiency or economy with a good system of forced lubrication.

A syphon worsted and lubricator box is shown in Fig. 340. The oil cup contains an internal pipe, in connection with an external cock and short pipe. The oil falls into an open-topped funnel-mouthed pipe, which is led down to the part to be lubricated. This pipe should have a gradual fall, without pockets. The worsted is weighted at the end resting on the bottom of the cup to prevent its being drawn into the internal pipe, and the distance the worsted extends down the pipe is regulated by a twisted wire, which is hooked over the top of the pipe and fastened to the end of the worsted. The end of the worsted inside the pipe should be just below the bottom of the cup. The rate of flow through the worsted can be regulated both by varying the height of oil in the cup, and by varying the number of strands of the worsted.

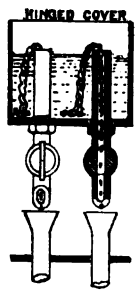


FIG. 340.—Syphon Worsted Lubricator.

Internal Lubrication.—Internal lubrication is unnecessary when condensed or distilled water is used for feeding the boilers in all unjacketed engines, and can be reduced to a very small amount when jackets are in use. Generally no internal lubrication is used for the engines of warships, but a moderate amount of oil finds its way into the cylinders, slide chests, etc., from the working rods and glands. If the air is excluded as much as possible, the working surfaces do not appear to suffer from corrosion in harbour.

Economy by Good Management.—It might be pointed out here that better adjustment, efficient setting of valves and gearing,

and good management and training, have enormous influence on the economy of fuel and repairs. Within personal experience, cases have been noticed where increased speed has been obtained, *on less I.H.P.*, in a comparatively old vessel; and in consequence the "steaming distance" has been increased. In large ships with a well-organised and well-trained staff, accustomed to the vagaries of the special machinery with which they have to deal, it should always be possible to obtain the "specified full power"; and under favourable conditions of weather and clean bottom, that power should give the maximum number of revolutions per minute originally obtained on the contractor's trials. A system of changing nearly the whole of the engineer staff in warships, on re-commissioning at intervals of two years, is not calculated to obtain good results. A better system suggests itself of changing one-third or one-half of the numbers of every rank and rating at annual intervals. The present system is productive of a very low mechanical efficiency of the machinery with which a warship is packed, and it appears absurd to perpetuate this old custom of entire change of crews in ships bristling with mechanisms of every sort and description, and with almost human differences of form and constitution. No organisation or previous experience (other than that of a highly trained engineer) can make up for the lack of training *necessary in the actual ship* for junior officers and men. What the nation requires of a 25-knot ship is a speed of 25 knots at all times, and this is only to be obtained by a crew accustomed to the actual ship.

CHAPTER XXXIII

THE ENGINEER OF THE WATCH

IN the **organisation** of the engineering staff three officers[1] are principally concerned—viz. (1) the Chief Engineer, (2) the Second Engineer, and (3) the Engineer of the Watch. The relative positions and responsibilities of these three officers is exactly similar to those of the captain, the second in command, and the officer of the watch on deck; but with the material difference, that the "Deck officer" can be relieved of his duties and responsibilities almost instantly, whereas the "Engineer of the watch," who has duties in connection with several water-tight compartments, cannot be properly relieved in many instances in less than ten or fifteen minutes.

The Chief Engineer is responsible for everything in connection with his staff and for the efficient working of the machinery and boilers, and the supply of hydraulic and electric power, as well as all mechanical fittings throughout the ship. In this duty he is assisted by the Second Engineer, who, acting as the Chief's executive officer, details all ranks and ratings *by name* for their various duties, and, when necessary, makes arrangements for meeting the requirements of the second in command of the ship. No matter how well the organisation may be ordered by the Chief, its efficiency and harmonious working are thus largely dependent on the tact and ability of the Second Engineer in carrying it into effect. He is thus generally the hardest worked officer in the ship, especially

[1] The general terms, Chief Engineer, Second Engineer, etc., are used as they are applicable to all vessels fitted with steam or other power. In large warships the Chief Engineer is usually an Engineer Commander; the Second Engineer is usually an Engineer Lieutenant, and officially known on board as the Senior Engineer Lieutenant; and the Engineer of the Watch is generally of Engineer Lieutenant's or lower rank. In small vessels, the Engineer of the Watch may be an Engine-Room Artificer (E.R.A.) or a Mechanician; but the actual naval rank does not affect the organisation, which must be the same in all vessels.

when the staff is a large one and only partially trained. He should keep his Chief thoroughly acquainted with everything which comes under his notice, and also be particularly careful not to omit any detail connected with the organisation—such as changing the duty or station of any officer or man, reporting men for insubordination, etc.

Neither the Chief, nor the Second, nor any officer should give an order affecting the organisation or routine without immediately acquainting the Engineer of the Watch at sea, or Engineer of the Day in harbour. The Engineer of the Watch (or day's duty, as the case may be) should always remember that he is the Chief's representative, and that being placed in that position of responsibility by his Chief, he virtually represents the Captain, who is responsible nominally for everything that goes on in the ship, although the Chief Engineer relieves him in greater part of the responsibility of the engineering department. The assumption of authority by the Second Engineer or other officer senior to the Engineer of the Watch does not relieve the Engineer of the Watch of his duty or responsibility, unless such senior officer gives him a direct order to that effect; in which case it becomes the first duty of the senior officer to inform the Chief Engineer immediately of what he has done and his reason for doing it. The King's Regulations distinctly define the responsibility of the Engineer of the Watch (see Arts. 972 and others), and he alone is responsible for carrying out all orders received from the Deck Officer of the Watch (who represents the Captain) or from his Chief Engineer.

The same organisation applies to each separate compartment, which during a watch or working hours is always placed in charge of the senior officer or petty officer detailed for that compartment, who, in consequence, is virtually the central pivot about which the organisation moves in that compartment.

Generally the Second Engineer is responsible for the discipline and cleanliness of the staff and department, and in addition he usually details the men for their various duties from day to day. The order and manner in which defects, adjustments, refits, or examinations are taken in hand are always determined by the Chief, and this points to the absolute necessity of the Second and other officers (who are detailed for certain sections of the department) keeping the Chief thoroughly informed of everything of importance which comes under their notice.

A Watch and Station Bill (now Form S.250B., but first introduced by the author about 1891) is prepared in which every officer and man (whose name should not occur more than once) is detailed for his watch and station. It is also usual to make some routine orders and post them conspicuously in the engine rooms for general guidance and reference. These orders should be such that they can be enforced strictly, and are then a very material assistance in the training of the staff in their several duties. (A copy of the routine and other orders used in H.M.S. —— is given at the end of this chapter; but, of course, these orders may not be entirely applicable to other ships.)

Training is allied to organisation and is also complementary to it, but it is quite easy for one to be efficient regardless of the other. The organisation details a petty officer or man to a particular watch or station, but by training alone can he become familiar with his duties when on watch or at his particular station. With a properly educated engineer, experience in one ship will generally give him sufficient training to carry out his duties in another; but with the men it is a different matter. Men require training in the actual ship until they become accustomed to the various peculiarities and differences in construction and working; and these remarks generally apply to the junior artificers as well as petty officers and men.

Inspection before Lighting Up.—Before lighting up, the boilers should be inspected to see that the water is at the correct height. On no account should a fire be lighted in any boiler unless the water is actually shown at a safe level in the gauge. For water-tube boilers, a height of about 1 to 2 inches above the bottom of the gauge is sufficient; because the water expands when heated, and, by the time steam is up, about a proper working height is shown. The correct working level is generally just above a half glass. With the larger tank boilers, a rise of about 2 to 4 inches is obtained while raising steam.

The engines should be carefully inspected to see that they are all clear, with the turning gear out and no obstructions in the crank and eccentric pits. All valves should be worked and closed again if necessary, but not jambed hard on their seatings. The lubricators should be inspected to see that they are clear, and worsteds properly fitted.

Warming Pipes.—In putting steam on to any part of the ship, it is very necessary that there should be an outlet for the water

which has condensed in the pipe, and the drain cocks should be opened first to allow this water to escape. If water and steam meet in a pipe, there is always danger from water-hammer. The action which takes place is somewhat as follows: the steam on first meeting the water is condensed, and a partial vacuum is formed near the surface of the water; the vacuum greatly increases the velocity of the incoming steam; the momentum increases with the velocity, and the increased force of impact, in combination with the stress caused by difference of expansion, may fracture the containing pipe.

When putting steam on to a cold pipe, before opening out further, about one minute should be allowed for each yard of pipe to be warmed, from the time when steam can just be heard moving in the pipe.

Internal feed pipes are particularly subject to water-hammer, and are frequently found split, especially if fitted near the working water-level.

Time required for raising Steam.—Whenever it is possible, a standard time should be agreed on between the Captain and Chief Engineer for raising steam; this allows many small repairs to be done which would otherwise be left for another opportunity. With *water-tank boilers*, it is not advisable to raise steam in less than six to eight hours. With *water-tube boilers*, the time in which steam can be raised, *in emergency*, is only limited by the time required to properly warm the engines through. Any undue hurry in the process of warming is likely to cause defects from unequal expansion of the various parts, and the time generally required may be safely put at about three hours.

There are few examinations of machinery which cannot be made in two working days, although repairs may take any time. The disposal of a certain amount of necessary routine work allows the engineer freedom to undertake more responsible repairs when the opportunity offers, and thus a fair standard of notice always contributes to efficiency.

Opening out and warming through Engines.—If the full pressure of steam be admitted to the cylinders or steam pipes before they are properly warmed, there is great danger from unequal expansion and consequent fracture. The engines should be warmed through gradually, and when possible, as soon as fires are lighted, the hot air and steam should be allowed to pass into

the steam pipes and engines. With water-tube boilers, it is sometimes necessary to keep the bulkhead or sectional valve closed until a feed pump can be started; otherwise the engines, taking away the steam so quickly, leave the water below the proper working level.

The jackets should always be used for warming engines, and the jacket drains opened as necessary. The cylinder and slide-chest drains should be kept open until steam is emitted to the bilge, and then opened occasionally afterwards. If the drains lead into the condenser, they may be left open until after the ship is under way. The engines should be drained when stopped for only a few minutes, so as to prevent any accumulation of water.

The circulating engine should be warmed up with the main engine, and the inlet and outlet on the condenser opened before the condenser is warmed by the steam which leaks through the engines. If only one condenser is required, see that the working one is connected with the eduction pipe, and that no steam can leak into the idle one. As a precaution, the inlet and outlet for every condenser might be opened, and kept open, while under steam.

The hand reversing gear should be tried while raising steam, to get the men into the way of knowing their stations. Start the circulating engine as soon as there is sufficient steam, usually from 15 to 20 lb. The condenser must be kept clear of condensation water by pumping out, if necessary.

When opening out, start at the condenser and retrace the path of the steam from the boilers, in order as below :—

Open drains to cylinders and slide chests; H.P. and M.P. to bilge until steam is emitted; and the L.P. drain to condenser until under way.

Open drains to jackets and steam to jackets.

Open connections for working auxiliary engines, as required.

Open manœuvring valve, about one-half turn generally.

Work main regulating valve and nearly reclose, leaving it very slightly open, so that it is not likely to jamb from difference of expansion.

Open emergency, or quick shut-off, valve.

Open separator drain. If worked automatically, this may be unnecessary.

Open main, or intermediate shut-off, valve; only a moderate amount, as necessary for speed ordered.

Open cross-connecting valve between engine rooms.

Open bulkhead valves as requisite; a moderate amount for speed ordered.

Open boiler steam valves, and notice that the valve is actually moved back.

Warm through the reversing engine, and as soon as steam rises to about 20 lb., sufficient to work the engine, the link should be run over and left alternately "ahead" and "astern" until the engines become lively. The links can be left over for five to fifteen minutes, according to the pressure and readiness of some one to check the engines.

While raising steam the capstan (for weighing anchor) and steering engines and gear, telegraphs, gongs, steam syren, and other small details required for getting under way, should be prepared, and tried ready for the time ordered.

The funnel guys should be slacked before lighting up, and inspected soon after getting under way, and readjusted if necessary.

Try Engines.—After setting the reversing gear, all-round type, to run steadily, turn a little steam on the engines. The cranks should begin to move a little after about five minutes of this treatment. Then turn on a little more steam until a revolution is made in each direction, as the link runs over. Then stop the link in the "ahead" and "astern" positions, and get about three revolutions in each direction, and continue until a vacuum of about 15 inches is formed in the condenser.

When trying engines, the auxiliary starting or pass valves should not be used to force any crank over the dead centre. Time should be allowed for the evaporation, or drainage, of the water from the cylinders and slide chests. At other times, when the engines are properly warmed, obey the telegraph promptly, because no harm is then probable from using pass valves.

Standing by.—When sufficiently warmed, steam may be shut off the engines, but not from the jackets. The links should be run over occasionally to clear the engines of water, the drains should be opened as necessary, and a little steam occasionally blown through the starting or pass valves to free them of water and keep the cylinders warm.

Under Way.—As soon as the order is given to proceed at a certain speed of revolution, the telegraph is put at "Half speed ahead." Set the link at the correct cut-off first, and then regulate

the number of revolutions and the steam pressure in the jackets which are to be used, and shut off the cylinder drains as necessary. For the jackets in use, shut off the drains and allow the water to accumulate until about half a glass is showing in the water gauge on the jacket-drain vessel, and blow down as necessary to keep it at about this level.

Inspect all lubricators to see they are working properly.

Linking-up.—Full advantage should be taken of linking-up by the main and separate links, to get economy and smooth running. The exact amount of linking-up for a certain speed and power is found by experience, and from careful observation of the indicator diagrams obtained and the corresponding ranges of temperature and coal consumption.

Knock on Bearings.—A knock can sometimes be mitigated by an alteration in the separate linking-up of the various link motions to vary the powers in the several cylinders. Hammering gradually increases the maladjustment, and the knock increases unless it is taken up at once. If the link fails to stop the knock, a little water and soft soap assists in forming a lather to fill the slack space between the rubbing surfaces. A knock can always be localised by turning on the water service to each bearing in turn.

Hot Bearings.—If the bearing is only slightly overheated, and cannot be cooled by the ordinary water service, some powdered black-lead is useful, if it can be got into the bearing through the oilways. For more serious cases, sulphur is used until a new bearing surface is formed. After a bearing has once warmed up there is always a tendency to again heat up, and it is good practice to keep the water service on until an opportunity can be made to refit or readjust it. In using water service, care should be taken not to wash the oil from the bearing. The water service should always be prepared for immediate use before getting under way.

Stop Engines.—First shut off steam, then centre the link.

Reversing Engines.—If the telegraph moves from "Ahead" to "Full speed astern," run the links over to "Astern," and then regulate the steam; which generally means opening the valve wider until sternway is got on the ship. All naval engines will stand reversing at full speed without shutting off steam; but the shock of reversal produces leaks and defects, and it should only be carried out in this way at low powers or in cases of apparent urgency.

If the telegraph moves from "Ahead" to only "**Slow**" or "**Half** speed astern," ease the steam, and then run over the links.

If the engines do not immediately reverse, run the links over again with a little steam turned on, or use the pass valves correctly.

Turbine Engines—Raising Steam and getting Under Way.—The routine differs slightly from that usual with reciprocating engines, but the general principle is the same, viz. the course of the steam is traced backwards to the boiler.

Previous to lighting fires, the following routine should be carried out in order (referring particularly to the arrangement shown in Fig. 364):—

1. Take out turning gear and inspect shafting, etc., to ensure everything is clear for moving engines.

2. Open screw-down drain valves at both ends of cruising turbines, at ahead ends of L.P. main turbines, and at after end of H.P. main turbine. Open self-closing drain valves on both ends of *astern* turbines. (All the above valves are generally kept open in harbour.)

3. Open all necessary supply and delivery cocks on forced lubrication and water-service systems.

4. If auxiliary steam is available, start circulating pumps and air pumps; but if not, start them as soon as possible after steam is raised.

5. Open master steam valves to cruising turbines, and manœuvring and regulating valves to H.P. and M.P. cruising, and to H.P. main, turbines.

6. Open slightly manœuvring valves to *astern* turbines.

7. Open bulkhead steam valves and main steam and sectional valves, as necessary, on boilers and main steam service.

8. *After fires are lighted*, the hot vapour and steam are allowed to pass through *all* the turbines until a pressure begins to show in the boiler gauges. The regulating and manœuvring valves are then almost closed, so as to allow only 1 to 2 lb. pressure to show in each turbine.

9. About half an hour before the time for trying the main engines, steam is opened to the spindle glands to assist in maintaining a vacuum in the condensers and in warming the rotor spindles.

10. Start the oil- and water-service pumps, and, if the other auxiliaries are working properly, the main engines may be tried.

11. To *try main engines*, shut the master and regulating valves

of the cruising turbines and the regulating valves of H.P. main turbine; and centre the manœuvring valves of the astern turbines. The manœuvring valve of, say, the port L.P. ahead and port astern turbines may now be opened slowly for *ahead* running until the shaft makes a few revolutions; the manœuvring valve may be run over to *astern*, and after obtaining a few revolutions in that direction, the manœuvring valve should be again centred. The S.L.P. ahead and S. astern turbines should be similarly tried; and, if correct, steam may be opened to the cruising turbines to try them also.

12. To *try the cruising turbines*, open the master valves, and then open the regulating valve of the H.P. cruising turbine until the shaft rotates a few revolutions; if steam is kept open for about a minute, the H.P. exhaust steam will set the M.P. cruising turbine in motion, and as soon as this happens the H.P. cruising steam should be shut off.

If it is inconvenient to run the H.P. cruising turbine for so long as one minute (because the ship is secured to a jetty, for instance), the M.P. cruising turbine can be tried independently, after the H.P. cruising turbine has been tried, by closing the H.P. cruising and opening the M.P. cruising regulating valve, which should be shut as soon as the engine moves.

13. The *H.P. main turbine* can next be tried by opening the H.P. main regulating valve.

14. The closed (auxiliary) exhaust steam may be used in case of urgency for warming through the main engines, but this is not a usual practice.

15. *Precautionary measures* are necessary, because the manœuvring valves are slack in their casings to allow free movement when hot or cold; and thus to prevent leakage the master valves are usually kept shut except when steaming with the fleet, or entering or leaving harbour, or on any other occasions when it may be necessary to go astern suddenly with either one or both astern turbines.

16. When the engines are ready for such occasions (as above) and under control of the manœuvring valves, all screw-down drain valves are kept *open*.

With the H.P. cruising turbine under steam, all screw-down drain valves are *closed* except those at the exhaust ends of the cruising turbines, which are kept slightly open.

With the M.P. cruising turbine under steam, and the H.P.C.

not under steam, the drains on H.P.C. are kept open slightly—say one turn on wheel.

With both H.P. and M.P. cruising turbines *not in use*, the drains on each are kept slightly open (as above).

17. The steam pressure in the spindle glands should be only just sufficient to prevent air leakage reducing the vacuum. Unnecessarily high pressure should be avoided, as it causes undue wear.

HARBOUR ROUTINE AND AUXILIARY WATCH-KEEPING

1. The Engineer of the Day (or Watch) is responsible for the good order and discipline of the department during his period of duty, and in the absence of the Second Engineer he is to inspect all adjustments and parts of machinery, boilers, etc., before they are closed up.

2. He is to see that all orders are intelligently carried out; to report any trouble or defect; and to make all necessary entries in the E.R. Register.

3. He is to muster the staff on each occasion of turning to, and to personally see that they carry on with their work at once.

4. Stoker ratings are to turn to at 6.0 A.M. and 1.15 P.M., and cease work at 11.45 A.M. and 4.30 P.M., unless otherwise ordered.

5. E.R.A.'s and Mechanicians are to begin work at 7.30 A.M. and 1.30 P.M., and cease work at 11.30 A.M. and 4.0 P.M., unless otherwise ordered.

6. Cases of "Insubordination" are to be immediately brought before the Duty Lieutenant or Lieutenant of the (Deck) Watch.

7. The auxiliary watches are to be properly kept and relieved, and, assisted by a junior officer, the department is to be visited by the Engineer officer of the Day during the middle watches, between midnight and 4.0 A.M.

8. The auxiliary watch-keepers are not to be taken away from their duties.

9. The separators on the auxiliary steam and closed exhaust systems are to be blown through every hour, or oftener if necessary. When the closed exhaust is in use, the dynamo watch-keeper is to be informed before starting the capstan or other engine likely to throw water into the dynamo cylinders.

10. Fusible plug pistols (Belleville boilers only), water-gauge glasses, and packing rings are to be kept in each boiler room where fires are alight.

11. Naked lights are not to be taken within 20 feet of a newly opened bunker, until it has been ascertained that no foul gas is present.

12. Fires are to be cleaned every night during the first and middle watches; and tubes are to be swept every alternate night.

13. Coal is to be kept clear of ash-pits and ash-pit doors.

14. Ash-pits are to have about 1 inch of water in them at all times (applies to water-tube boilers), and are to be kept clear of ashes.

15. Draught plates are never to be unshipped or opened wide.

16. Furnace doors are not to be secured open when firing.

17. Furnaces are to be fired lightly—about three shovelfuls at a time, at regular intervals and in proper rotation. The fire should be 5 to 6 inches thick, and level.

18. In case of serious accident, such as a burst or overheated tube or loss of water, etc. :—1st, close all smoke-box, furnace, and ash-pit doors ; 2nd, open safety valves (slowly at first) ; 3rd, close steam stop valve ;—send a messenger to report to Engineer Commander at once.

19. Safety-valve gear of boilers in use is to be worked every forenoon, and the valves lifted when raising steam, and every Monday afterwards.

20. Water and pressure gauges are to be kept clear, and the wire-pulls kept in good order and free, and they are to be tried once in each watch.

21. Feed check valves, and steam valves to feed engines, are never to be closed without special order.

22. When starting feed, fire, and drain tank engines (of the Weir type), the by-pass handles are to hang "down" ; and when fairly under way, the handles are to be raised ; and care should always be taken that the *turning gear is out.*

23. Drains on slide casings of feed and fire engines should be opened occasionally to clear away any water.

24. A "stand by" feed engine is to be tried once in every four-hour watch, and afterwards logged in E.R. Register.

25. Any difficulty with feed pumps or automatic regulators is to be at once reported.

26. Heavy mineral oil only, and as little as possible, is to be used on the piston and slide rods of auxiliary engines.

27. For *feed* engines (Weir type only), heavy mineral oil is to be used as follows :—half a turn of ram every *two* hours ; for *fire* engines, half a turn every *four* hours. The steam valve to lubricator is to be opened only when forcing oil into the slide casing.

28. Automatic feed handles are to be held up when "blowing down."

29. Boilers are to be blown down once in each watch when using *shore* water, and at 8.0 A.M., noon, and 4.0 P.M. when using *distilled* water.

30. Any density in the boiler water is to be reported at once, when definite orders will be given as to blowing down.

31. The water-level in each feed tank should be kept above 3000 gallons, but not above 4000 gallons.

32. Reserve feed tanks are to be sounded carefully when taking over or relieving the watch.

33. 2 lb. of lime, thoroughly mixed with water, are to be put into the lime tank at 8.0 A.M. and 8.0 P.M.

34. Lime tank valves are to be kept slightly open, and to be opened about one turn for about one minute every half-hour, unless otherwise altered.

35. Except when under repair, the after fire and bilge engines are to be kept for *fire-main* purposes, and the forward engines for the *bilges.*

36. Before using any pump for the fire-main, it should be washed through by pumping overboard for a few minutes.

37. For ordinary purposes, 20 lb. pressure is to be kept in the fire-main ; but at fire or general quarters, the pressure is to be increased to 30 lb., using two pumps if required.

38. All bilges are to be kept pumped out, and No. 1 boiler room should be specially watched.

39. The auxiliary *closed exhaust* is to be used for the evaporators, and a pressure of at least 20 lb. is to be maintained in it. The relief valves are to be frequently worked, but not held open.

40. When the evaporators are at work, the water passing through the auxiliary air pump is to be *tasted* every hour, and it is to be *tested* by "chemical test" in the middle, forenoon, and first dog watches.

41. When distilling for ship's use, the flow of water is to be gauged twice during each day watch and once in the middle watch (during the visit of the E.O. of the day).

42. The density in the evaporators is not to exceed 20° ($\frac{2}{32}$), and any signs of priming are to be at once checked and reported.

43. Running the after dynamos should be avoided as much as possible when the closed exhaust is in use. The separators are at all times to be kept clear of water, and the engine cylinders drained promptly ; the dynamo engine should be eased down if necessary, and if the after dynamos are in use, ring up the E.R. and ask for the separators to be cleared, and, if necessary, for the closed exhaust relief to be opened.

44. Dynamos are not to be run above their maximun speed. Bad sparking at the brushes, or excessive heating of the armature or leakage of current, is to be reported at once.

45. Mineral oil only is to be used in the crank chambers, and the oil pressure gauge is to be carefully watched.

46. Olive oil only is to be used for lubricating the tail bearing of the armature spindle.

47. The chief stoker of each night watch is to visit the dynamo room to see that all is correct, on being relieved.

SEA ROUTINE

1. The general instructions contained in the Steam Manual are to be carried out.

2. All bunker doors not in use are to be kept closed, and those in use are to be kept clear for closing at any time.

3. All inner and double-bottom compartments are to be closed before the time ordered for getting under way, and the condition reported to the Engineer Commander.

4. The special instructions for steaming Belleville boilers, and their care and maintenance, are to be carefully read and carried out. (Circular Letter No. 121 of 1906.)

5. Steam is not to be admitted to main or auxiliary steam pipes until all water has been drained out, and then only very slowly. Care is to be taken that the drains are clear and not choked.

6. Separators, main and auxiliary, are to be kept clear of water, and the blow-down valves properly closed when not in use. The silent blow-down valves are to be opened very slowly, and their use avoided as much as possible. The blow-down and silent blow-down valves are not to be opened by any one below the rating of Leading Stoker.

7. As a general rule, three hours are required for warming through the main engines, and engines should be ready for trying fifteen minutes before the time ordered for getting under way.

8. Steam in boilers is to be kept at about 250 lb., and not allowed to fall below 175 lb. in the engine room.

9. Increase of steam pressure is to be checked by opening:—1st, smoke-box doors on idle boilers in connection with the same funnel; 2nd, economiser doors of boilers in use; 3rd, smoke-box doors of boilers in use.

10. Smoke-box doors are not to be opened beyond the locked position, unless it is for the purpose of renewing a fusible plug, when care is to be taken before opening that the boiler is in a safe condition.

11. Air blowers are to be worked at a pressure of 10 lb. for ordinary steaming; 13 lb. for $\frac{3}{5}$ power; 15 lb. for $\frac{4}{5}$ power; and 17 lb. for A.N.D. power.

12. When F.D. fans are working, the inlets from the upper deck are to be kept open, and the flaps below the fan casing closed.

13. Feed water is to be tended by E.R.A.'s or Mechanicians on watch in boiler rooms.

14. Klinger water gauges may be shut off if leaking, and are to be reported to E.O. of the watch, who will have new glasses fitted as soon as possible; meanwhile, they are to be opened every half-hour to check the height of water. (See also Art. 20, Harbour Routine.)

15. Where there is no density in the boiler water, the litmus test for boilers and feed tanks and the chemical test for feed-water and air pumps are to be made at 9.0 A.M. and 10.0 P.M., unless otherwise ordered.

16. Arts. 9 to 11, 13 to 32 of Harbour Routine are to be generally complied with.

17. Ashes are to be reported to the bridge at 5 Bells in each watch, and the average revolutions every hour.

18. Lime is to be put into the tanks in each watch, the quantity as ordered by the Engineer Commander. Valves on tanks are to be opened every fifteen minutes, or as ordered.

19. At all powers, the L.P. separate links are to be kept in full gear.

The H.P. separate links are to be set at 4 for 105 R.P.M. and below.

" " " " 2 for 100 " "

" " " " 0 for 120 " and above.

The M.P. " " " 4 for 110 " and below.

" " " " 2 for 120 " and above.

The main links are to be linked up from 50 per cent to 75 per cent, so as to keep the pressure in H.P. receivers as high as possible; but the pressure should not exceed 240 lb. at full power.

20. The cylinder steam jackets are to be used for warming through engines, but, unless otherwise ordered, are to be shut off as soon as the engines are under way and revolution telegraph set. The steam traps on H.P. and M.P. jackets are to be shut off when not in use.

21. Main piston and slide-rod glands are not to be screwed up without permission from Engineer Commander.

22. The E.O. of the watch is to carefully turn over to his relief the exact condition of the bearings, mentioning any cases where the water service is being used more than ordinarily.

23. Internal water service is *not* to be opened at any time to main bearings or plummer blocks without special orders.

24. Water service is not to be put on, or taken off, without orders from the E.O. of the watch, except in urgent cases, when it is to be reported at once to the Engineer Commander.

25. In general, the oil in lubricator boxes is to be kept from $\frac{1}{4}$ to $\frac{1}{2}$ inch below the top of the tubes, and every endeavour is to be made to prevent waste.

26. Ordinary mineral oil is to be used in slide and piston rod *glands*, and heavy oil for piston and slide *rods*. No oil is to be used for *internal* lubrication of main or auxiliary engines without orders from Engineer Commander.

27. *Worsteds* for crank-head bearings are to be used as follows :—

For 65 revs. and below— Nos.	2	5	8
65 ,, to 80 . . . ,,	2	4 . 6	8
80 ,, ,, 95 . . . ,,	2	4.5.6	8
95 ,, ,, 105 . . . ,,	1 . 3	4 . 6	7 . 9
105 ,, and above . . ,,	1.2.3	4.5.6	7.8.9

28. The level of mineral oil and fresh water in the thrust boxes is to be maintained by adding fresh water from time to time as required. The suction strainers are to be kept clear, and the temperature of the mixed oil and water should not exceed 150° Fahr. (N.B. This refers to special apparatus described in Chapter XVI.)

TELEGRAPHS AND TELEMOTOR STEERING CONTROL GEAR

All orders will be received through the telephone in starboard engine room, and the Engineer officer of the watch is to see that they are at once carried out.

The engines are at all times to be worked to correspond with the orders shown on the engine-room telegraphs, unless special orders are given to the contrary.

The telegraphs can be connected up *correctly* in *any* position, and it is unnecessary to put them to *Stop* when changing over from the *fore bridge* to the *conning tower*, or *vice versa*. *Before disconnecting*, they are to be first connected with *both* positions, so that the engines are at all times under telegraph control. When the change is complete, it is to be reported from the forward ammunition lobby to the *conning tower* (this being the only means of communication), and the telegraphs tried from the new working position.

When changing over the steering control gear from *fore bridge* to *conning tower*, or *vice versa* :—

The telemotor control wheels must *both* be in the *mid position*; and after the change is made the rudder, with the permission of the Officer of the watch, is to be run hard over each way, port and starboard, and, if correct, reported connected with the new working position.

When the steering gear is *disabled*, the E.O. of the watch will be informed, and a definite number of revolutions telephoned for the *port engine*; and the *starboard engine* is to be worked according to the *revolution telegraph*. For extreme speeds, slow or full, the *engine telegraphs* will be used for both port and starboard engines.

When ordered to connect up *hand-steering gear*, the senior Engine-Room Artificer (starboard engine room) and a Stoker Petty Officer (port engine room) are to be sent to the steering flat, and they are to report to the conning tower "Ready to change over to the hand gear." Nothing is to be done

until the order is received to "Carry on": when the brake is to be put on; the hand wheel set to correspond with the rudder; the clutch put in gear; and the brake thrown out; and, all being in order, the change is then to be reported to the conning tower.

ORDERS FOR RAISING STEAM

H.M.S.....

The auxiliary watch-keepers are to call:—

Engineer Officer of Watch at .....M. Mr. .....
Steaming Watch at .....M.
Senior Engineer Lieutenant at .....M.
Light fires in boilers ..... and connect by .....M.
" " " " " "
" " " " " "
" " " " " "

Full watch to be at their stations by .....M., when commence warming through engines by routine, seeing that all turning gears (main engines, feed pumps, etc.) are out or removed; all exhaust sluices, and necessary circulating inlets and outlets, drains, etc., are open.

Take off closed exhaust and start ..... auxiliary and ..... main circulators.

Rig windsails, Nos. ..... by .....M.

Put in all worsteds at .....M.

Boilers to be connected up as above; stern glands eased back, and main engines thoroughly warmed through and drained

ready for trying at .....M.
and for getting under way at .....M.
Main boiler stop valves to be opened ..... turns.
Bulkhead valves to be opened ..... "
Intermediate steam valves to be opened ..... "

Open bunkers, Nos. .....port, and ..... starboard.

Start Nos. .....blowers at .....M.

..... steering engine and ..... telemotor to be reported to E.C. at .....M.

Forward capstan engine to be reported to E.C. at .....M.

Funnel guys to be slacked back, sirens and telegraphs to be reported to E.C. at .....M.

Observe carefully precautions on opening out steam pipes and water-hammering while warming through, and about naked lights on opening bunkers.

The Engineer of the Watch is to examine all anchor bolts of boilers lighted up, and to log the result in the E.R. Register.

These orders to be read and initialled by officers concerned.

(Signed) ..... *Engineer Commander.*

PART X

RECENT DEVELOPMENTS

CHAPTER XXXIV

MARINE STEAM TURBINES

Introductory.—Turbines may be divided into three classes according to the manner in which the steam causes their rotation. These classes are:—

(1) *Impulse, action,* or *velocity*; of which the most notable examples are the Pelton wheel (water), De Laval (modern steam type, Fig. 341), Rateau, Zölly, and many others of lesser note.

(2) *Reaction* or *pressure,* of which there are no examples in modern practice; but Hiero of Alexandria invented this type about 130 B.C., and it was used by De Laval many years ago in the form shown in Fig. 342.

(3) Combined *impulse* and *reaction* (Fig. 343); of which the most notable are the Parsons and the Curtis types. The Curtis type is frequently classed as an "impulse" turbine; but, as will be seen later, there may also be reaction, although not to the same extent as in the Parsons.

There is a further classification of turbines which, being influenced by their design and construction, depends on the direction of flow of steam with reference to the turbine axis. The names are self-explanatory: parallel or *axial* flow, which includes nearly all marine turbines; *inward radial* flow; *outward radial* flow; and *mixed* (axial and radial) flow. Generally axial flow is best suited for steam and elastic fluids; and radial, or mixed, flow for water and inelastic fluids.

All turbines work on the principle of conversion of kinetic energy into work, and by this means rotating a turbine wheel or a series of wheels. The kinetic energy (energy of motion) is usually produced from some form of potential energy, such as hydraulic pressure or the heat energy of either air or steam or other gas.

As an example, consider a jet of water issuing from a horizontal pipe and striking a wall at a little distance from the mouth of the pipe or jet (Fig. 344). If the weight of water delivered per second

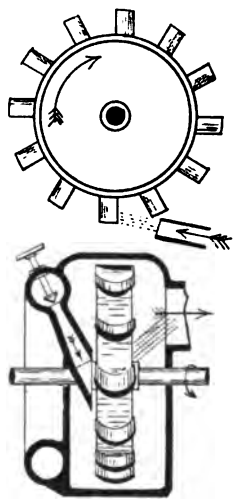


FIG. 341.—"Impulse" Turbine.

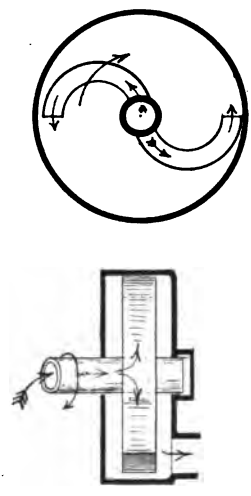


FIG. 342.—Reaction Turbine.

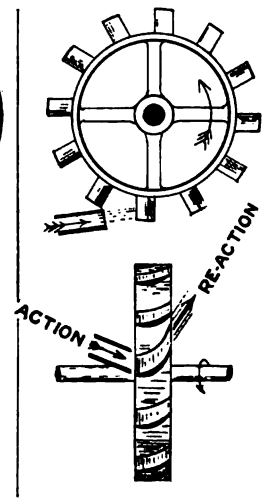


FIG. 343.—"Impulse" and Reaction Turbine.

be W, and its velocity V in feet per second, then the force of impact or thrust or push on the wall, when stationary, evidently—

$$\text{Force of impact, or thrust} = \frac{W \cdot V}{g}.$$

Next, suppose that the wall is pushed away from the mouth of the pipe and moves with a velocity, U; and further, that the pipe being fixed, a continuous succession of walls present themselves in front of the jet, similarly to the arrangement shown in Fig. 341. Then the change in velocity of the jet

$$= V - U.$$

$$\text{Thrust} = \frac{W}{g}(V - U).$$

Work done per second = thrust × distance moved in one second

$$= \frac{W}{g}(V - U) \times U.$$

From which the

$$\text{Horse-power (per minute)} = \frac{W \,.\, U \,.\, (V - U) \times 60 \text{ (seconds)}}{g \times 33{,}000}.$$

Condition for Maximum Efficiency.—It is evident, for the condition of maximum work transferred from the jet to the wall or blade, that the jet should be moving in the same direction as the blade at the instant of striking, and the fluid should not be dispersed over its surface but concentrated, so that every particle of its energy should be entirely absorbed without change in its direction

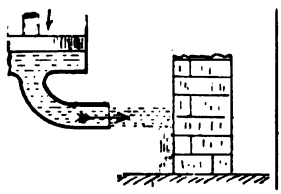


FIG. 344.—"Impulse" exerted by a Jet.

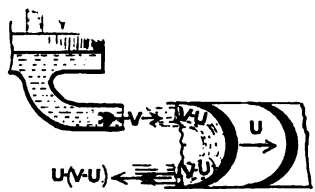


FIG. 345.—"Relative" Velocity of Jet and Blades.

of motion at the instant of parting with its energy. This condition may be very nearly obtained by some such system as shown in Fig. 345; in which the blade is curved in the form of a semicircle, and the jet strikes so that the stream is exactly reversed in direction by the blade.

The *relative* velocity of the jet to the blade at the instant of striking = V − U; and, neglecting frictional resistances, this velocity is maintained during its motion until the stream reverses its direction and leaves the blade. Thus, at the instant of exit from the blade, the stream has a relative velocity = −(V − U): because during its passage between the blades the velocity, being reversed in direction, changes its sign from positive to negative. The *absolute* velocity (relative to the earth) of the stream after exit evidently

$$= U - (V - U) = 2U - V.$$

If the whole of the kinetic energy of the jet be converted into

work in rotating the wheel, the stream must have no absolute velocity on its exit from the blade; and thus—

$$2U - V = 0, \text{ or } U = \tfrac{1}{2}V.$$

In other words, the condition necessary to obtain the maximum quantity of work from a jet of fluid is that the velocity of the blade must be exactly one-half that of the jet; this is, therefore, the condition necessary to obtain maximum efficiency.

In practice, there are certain losses which prevent this maximum from being obtained; and, stated briefly, these losses in steam turbines are caused by:—The stream not flowing into the blade passages exactly at right angles to their axis of rotation; skin-friction of the steam coursing at high temperature through the nozzle and blade passages; unavoidable leakages through the clearance spaces; eddy currents due to insufficient blade velocity and errors in workmanship; some velocity, and therefore energy, must remain on exit from the final blade passages so that the flow may be continuous; and the exit stream does not leave the blade passages exactly at right angles to their axis of rotation.

In the "impulse" type, the steam (or other fluid) issues from a jet with great velocity and acts on a series of blades or sails standing out radially from a wheel which is revolved by the "action" of the steam. The nozzle is formed internally, as shown in Fig. 346, so that its area at first rapidly converges (corresponding to the *vena contracta*) to about 58 per cent of the original area, and then diverges to areas corresponding with the adiabatic expansion of the steam. The action inside the nozzle may be understood by imagining a bubble of steam to just fill the space at the smallest diameter of the nozzle. The bubble moves forward (like a piston) under the difference of pressure behind and before it. As it moves forward the pressure before it continues to diminish, because the steam expands to fill the rapidly increasing area; and under the influence of this continuous application of force, the bubble gathers increasing velocity and kinetic energy. By this means some of **the heat energy of the steam is transformed into energy of motion of the steam, and the pressure and temperature of the steam correspondingly decrease in exact proportion to the heat converted into kinetic energy.** Thus, in a very short, and also properly designed, nozzle it is possible for the steam at one end of the nozzle to have an absolute pressure of 150 lb. per square inch,

a corresponding temperature of about 359° F., and only a very slight velocity; while at the discharge end the steam may be at an absolute pressure of 1 lb. per square inch (corresponding to 28 inches vacuum, and a temperature of 102° F.), and move with a velocity of over 4000 *feet per second.*

The total heat energy of 1 lb. weight of steam at 150 lb. absolute pressure is roughly 1190 b.t. units, and some 40 per cent of this may be transformed into kinetic energy $= \left(\frac{V^2}{2g}\right)$. It is this

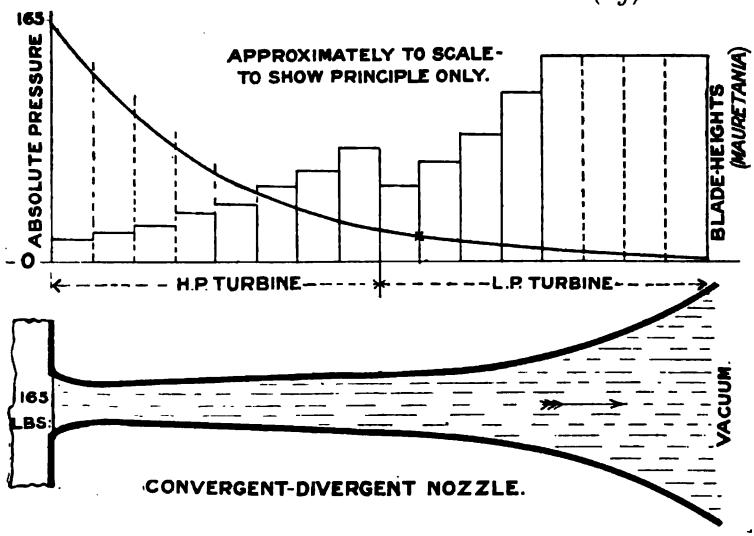


FIG. 346.—Convergent-divergent Nozzle.

energy of motion which acts on the blades or sails of the turbine wheel, and a further transformation of energy takes place in creating rotation of the wheel.

In the modern De Laval and other "impulse" turbines there is no appreciable drop between the pressure at the nozzle discharge and that after the steam has left the wheel. The drop in pressure takes place almost entirely in the nozzle, and the turbine wheel is rotated by the action of the steam moving with great velocity; this constitutes a *velocity* turbine.

In a *reaction* turbine the reaction of the (hydrostatic) fluid pressure in the nozzle causes, or tends to cause, the nozzle to move away in a direction opposite to that of the steam flow. A fall in pressure and temperature may thus take place both

within the nozzle (action) and within the moving blade passages (reaction).

A fall of pressure within the moving passages constitutes a *pressure* turbine, and by this name the Parsons type is sometimes referred to; but it is really more particularly applicable to water turbines of the Fourneyron and Thomson types, in which the fall in pressure takes place almost *entirely* within the blade passages.

Kinetic Energy of Steam.—In considering the conversion of energy which takes place in a properly constructed convergent-divergent nozzle, it is fairly evident that the expansion is approximately adiabatic; that is, little or no heat is added to or taken from the steam during its passage through the nozzle. This is particularly true of a short nozzle, such as fitted in De Laval and other turbines; but, due to radiation and other losses, is not quite true of a Parsons turbine in which the nozzle, made up of successive rings of blade-passages, is very long.

By neglecting any loss which may be occasioned by radiation, etc., and considering the expansion to be adiabatic, the law of the conservation of energy states that the total energy in each 1 lb. weight of fluid at one end of the nozzle must be exactly equal to that at the other.

If H = total heat in 1 lb. of steam at the high pressure end,
and h = „ „ „ „ „ low pressure end,

then, evidently, $$H = h + \frac{W \cdot V^2}{2g};$$

$$\text{or, kinetic energy} = H - h = \frac{V^2}{2g} \text{ for 1 lb. weight.}$$

If the steam at the inlet end of the nozzle is saturated steam, then, according to Professor A. Rateau—

Kinetic energy per 1 lb. weight (W = 1)

$$= J\ (T_0 - t_0) \left\{ \frac{L}{T_0} + \frac{C(T_0 - t_0)}{T_0 + t_0} \right\}$$

where J = Joule's equivalent = 780 ft.-lb.;
C = specific heat of liquid (water = 1);
T_0, t_0 = absolute temperatures;
L = latent heat of saturated steam at temperature T.

The Rateau formula is correct for initially saturated steam and

purely adiabatic expansion, but for practical purposes the value of h, being dependent on any circumstances (not necessarily adiabatic), should be determined. This is, however, almost impossible, because h varies with every engine under different working conditions.

As an example, suppose that 1 lb. weight of saturated steam at 180 lb. gauge pressure (temperature, 380° F.) expands in passing through the nozzle to an absolute pressure of 2 lb. (temperature, 126° F.), and that before entering the nozzle the steam is saturated, but that after expansion it is wet and contains 25 per cent of water; what is the velocity created by its expansion in the nozzle and its kinetic energy per 1 lb.?

$$H = 966 - 0{\cdot}7\ (380 - 212) + (380 - t);$$
$$h = \tfrac{3}{4}\{966 - 0{\cdot}7\ (126 - 212)\} + (126 - t);$$
$$H - h = 332{\cdot}7 \text{ b.t. units};$$
$$V = 8\sqrt{780 \times 332{\cdot}7 \text{ (ft.-lb.)}} \text{ in feet per second}$$
$$= 8\sqrt{259{,}506} = 8 \times 509;$$
$$\text{velocity, } V = 4072 \text{ feet per second};$$
$$\text{kinetic energy} = H - h = 332{\cdot}7 \text{ b.t. units}$$
$$= 259{,}506 \text{ ft.-lb. per 1 lb. weight of steam.}$$

If calculated by using the Rateau formula, the kinetic energy appears to be less (234,000 b.t. units about), which shows that 25 per cent of moisture may be too great a proportion to allow for condensation during adiabatic expansion in a properly designed nozzle.

De Laval Turbine.—This turbine is of the "impulse" type (Fig. 341), and the whole of the energy developed in the nozzles is imparted to a single rotating wheel and ring of blades. The number of nozzles may be one or more, and in recent types the nozzles are not arranged at equal intervals around the wheel circumference but are concentrated into groups of nozzles for the larger powered turbines. Fig. 347 shows a section through a De Laval nozzle, *B*. Steam is regulated by a valve, *A*, and enters an annular casing, *C*, from the boiler. The blades of the rotor wheel are concave towards the jet, and the horns of the cavity are in axial line with the wheel. The jet is necessarily fitted, when at the side of the wheel, at some angle with the axis other than a right angle, and is fixed usually at 20° (as shown) to the face of the wheel. A section of the rotor wheel is shown in the right-hand figure.

Fig. 347A shows a sectional elevation and a plan (with the gearing cover removed) of a 20-H.P. De Laval steam turbine driving a belt pulley, *M*, through gearing, *HJ*. The actual turbine wheel is shown at *F*, and is of very strong construction near

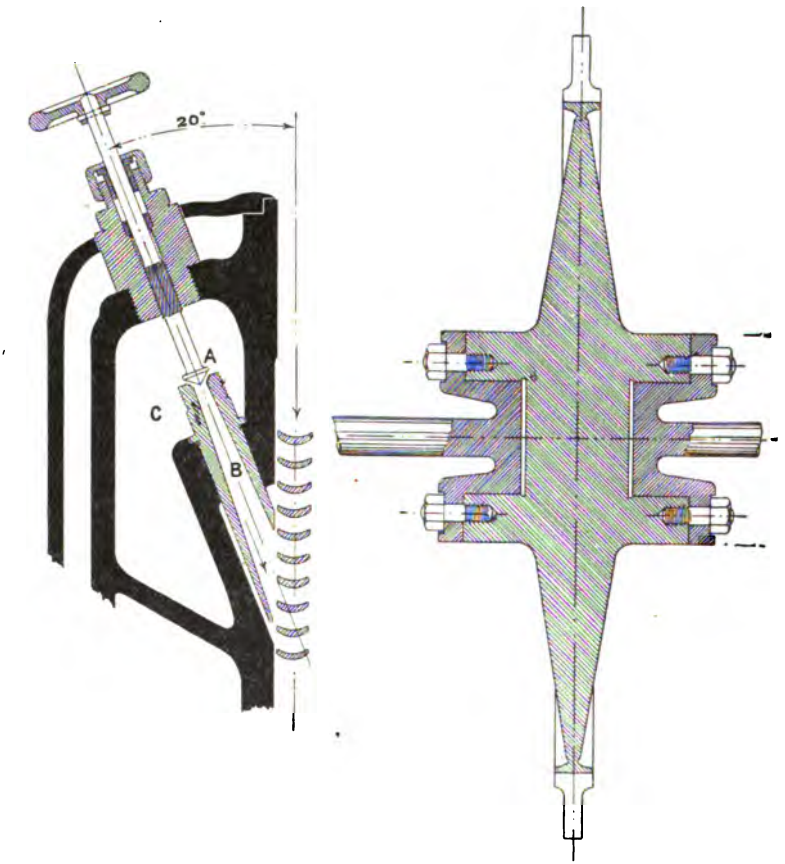


FIG. 347.—De Laval Nozzle, and Rotor Wheel.

the axis, so as to resist centrifugal action set up by the great rate of revolution of the shaft. This turbine, on account of its great speed of revolution, is unsuitable for ship propulsion; but it occupies very small space, and, due to its consequent small weight, it is usefully employed for dynamos, fans, etc., which naturally require a high speed.

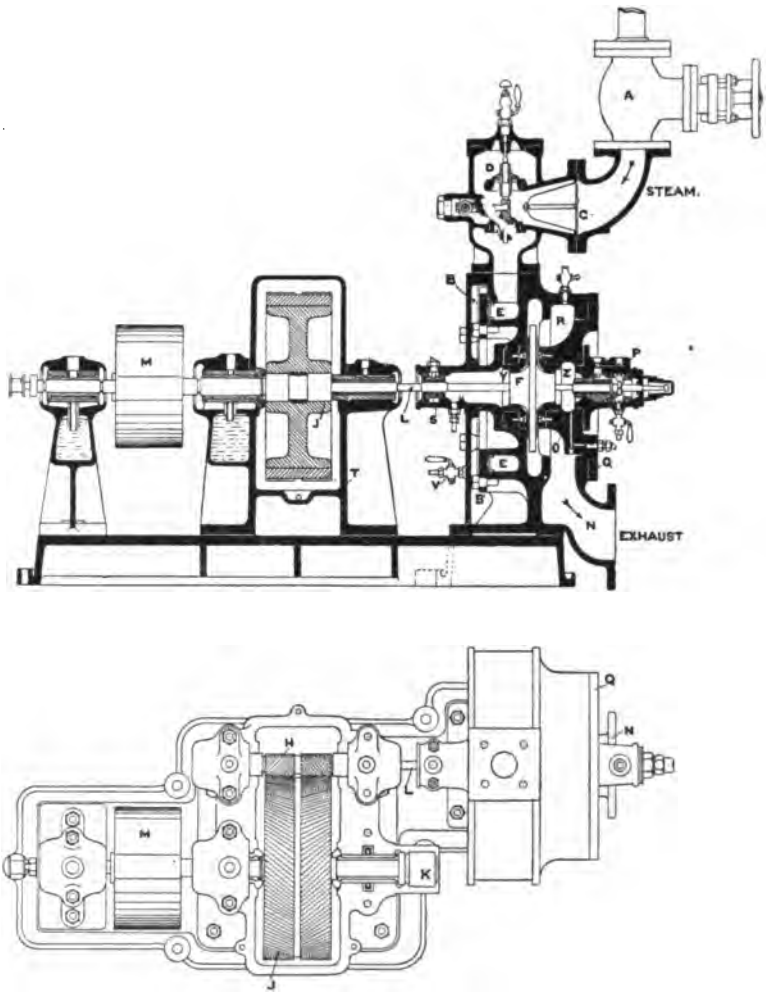


FIG. 347A.

A, Steam regulating valve.
B, Face-plate enclosing circular steam space, *E*.
C, Gauze-wire filter.
D, Governor-valve box and valves.
E, Annular steam space (*C* in Fig. 347).
F, Turbine wheel.
H, Pinion wheel on turbine shaft.
J, Gear wheel on driven shaft.
K, End bearing of gear-wheel shaft (enclosing governor).
L, Turbine shaft.
M, Pulley on gear-wheel shaft (for belt driving).
N, Exhaust.
O, *Q*, End casing, enclosing wheel and exhaust space.
P, *S*, Turbine shaft bearing and casing.
R, Exhaust space.
T, Gear-wheel casing.
V, Drain to steam space.
Y, *Z*, Casings for turbine wheel.

The following table gives some of the principal sizes of De Laval turbines constructed by the English De Laval Company :—

Brake Horse-Power.	Middle Diameter of Turbine Wheels.	Revolutions of Wheel per Minute.	Velocity of Blades.	Revolutions of driven Shaft per Minute.	Weight (about).
B.H.P.	Inches.	R.P.M.	Feet per Minute.		Cwts.
5	4	30,000	515	3000	3¼
15	6	24,000	617	2400	5½
30	8⅞	20,000	774	2000	11
		With two pulleys :			
50	11¾	15,000	846	1500	29
100	19¾	10,500	1115	1050	72
300	30	7,500	1378	750	161½

The general construction of the turbine may be gathered from Figs. 347 and 347A, and the reference to Fig. 347A.

Parsons Radial-flow Turbine.—There are two types of Parsons turbines—the radial, and parallel or axial, flow. The radial-flow type is not used for marine work, and seldom adopted for shore work, so that the parallel-flow type only is described in detail. The radial-flow turbine is similar in principle to the *inverse* motion of a centrifugal pump, and a flow of steam in an outward radial direction causes a rotation of the shaft. There are alternate concentric rings of fixed and moving blades, and the steam entering near the axis passes first through a ring of fixed blades which direct it on to the first ring of moving blades; it is then redirected by a second ring of fixed blades on to a second ring of moving blades; and so on, until the steam passes through a final ring of (moving) blades and escapes into the exhaust.

Parsons Axial-flow Turbine.—A great number of blades are fitted in rings on a hollow cylinder, which forms the rotating part of the turbine, and is called the *rotor*. Alternate with these rings of moving blades, which stand radially outward from the rotor, there are a number of blades fitted in rings inside an outer and fixed cylinder; and these fixed blades stand radially inward so that their tips nearly touch the rotating drum. The moving blades almost touch the fixed cylinder between the fixed blades. Looking at the tips of a few moving blades, shown in cross-section in Fig. 348, the course of the steam is shown by the stream lines from the first ring of fixed blades, shown in all-black section. It should be

remembered that the Parsons turbine is of the combined "action and reaction" type, and therefore the blades do not stand exactly in an axial line, but somewhat inclined to it, as shown. This inclination varies in different turbines, and in nearly all those recently constructed the inclination is gradually decreased for the last few rings near the low pressure or exhaust end. Referring to the diagram, it is seen that the first ring is fixed (fitted in the outer cylinder), and guides the steam so as to *act* on the first ring of moving blades (fitted on the rotor).

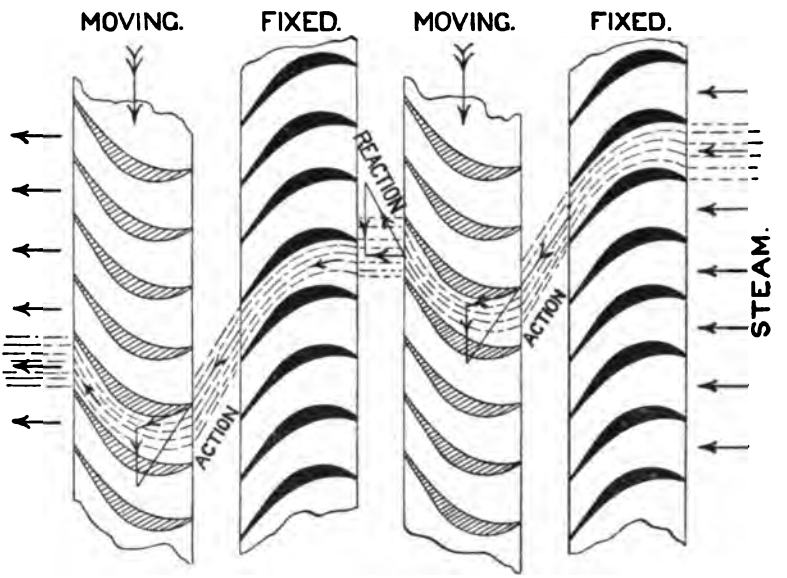


FIG. 348.—Course of Steam in Parsons Turbine.

A triangle of velocities shows the general action; the steam enters with absolute velocity, which becomes relative in the moving blade passages. The *backs* of the moving blades are shaped, on entry, so as to lie parallel to this relative velocity. The direction of motion of the steam changes within the passage between the moving blades, and the steam leaves the moving passage with a relative velocity which *reacts* on the blades. Immediately on leaving the moving passage, the relative velocity becomes absolute, and thus the steam moves in almost an axial direction into the second ring of fixed or guide blades, which are set so that the steam enters them without shock; the face of each guide blade is shaped so that on

entry the steam travels almost in an axial direction, and the change in direction necessary for "action" on the second ring of moving blades is effected almost entirely within the guide blade passages.

There are a large number of rings of blades in a Parsons turbine, and consequently there is only a small drop (seldom greater than 2 lb. per square inch) from fixed ring to fixed ring. The total difference in pressure between the inlet and exhaust produces the kinetic energy which constitutes the impelling force, and, the fall in pressure being divided into stages, only a proportionate quantity of the total energy is developed and utilised in each. Thus, compared with the development of the total energy in a single stage (as in the De Laval single-wheel turbine), the velocity of the periphery of the rotor and moving blades is moderate. This produces a moderate rate of revolution, and allows the application of a multi-expansion turbine to marine propulsion without speed-reducing gear between the rotor and the propeller.

Referring to the *ahead* turbine *K* in Fig. 356, it will be noticed that the blades are longer as the steam approaches the exhaust end. (Each of the sets of blades of equal length is called an *expansion*.) These expansions allow for the greater volume of the steam obtained by the fall in pressure as the steam expands and does work. A diagram of the relative blade heights for the *Mauretania* is shown in conjunction with Fig. 346. The H.P. turbine rotor is 96 inches in diameter, and the blades increase in length from $2\frac{1}{2}$ to 12 inches. The L.P. turbine rotor is 140 inches in diameter, and therefore the annular passage area being greater, the blades at the high-pressure end, where the exhaust from the H.P. turbine enters, only need to be 8 inches long. At the exhaust end the blade length is 22 inches, with a tip clearance of $\frac{8}{100}$-inch. The astern rotor drum is 104 inches in diameter.

Parsons Blades and Attachment.—The original Parsons turbines, from about 1887 to 1894, were fitted with flat blades and straight passages set at an angle of about 45° with the axis of the rotor. (An actual turbine of this type is in the South Kensington Museum.) Turbines of this description are reversible by changing over the inlet and outlet of the steam; but, steam being an expansive fluid, the economy during reversal is extremely low, and only a moderate proportion of the full-ahead power can be developed.

About 1894 the sectional form of blade was altered to that

shown in Fig. 348, which shows tips of blades of about the full size for a small or moderate sized turbine. This increased the ahead efficiency, but almost excluded the possibility of reversal.

Originally the blades were made of cast brass, but they are now cut the required length from drawn section wire. Each blade is a separate piece; and grooves are cut in the outer circumference of the rotor, and in the inner circumference of the fixed cylinder or case, for the reception of rings of blades. The moving blades stand radially outward from the rotor (as shown in Fig. 349), and the fixed blades stand radially inward from the fixed casing. Their root ends are fitted into the grooves with a *filling piece*, also cut from drawn section wire, between each pair of blades. The embedded parts of the blades and the filling pieces are driven home tightly against each other, and then caulked in at the lips of the grooves. The grooves in the rotor drum are slightly undercut or dovetailed, and the sides are left with serrated edges. The rigid connection of the blade and drum is necessary to resist the centrifugal action, as well as the impact and reaction of the steam. The method of caulking adopted by Messrs. J. Samuel White & Co. of East Cowes, of squeezing in the sides of each groove by roller pressure, is preferable to hammer caulking, which may tend to loosen the blades if carried to excess. The angle at which the blades are set in relation to the turbine axis depends on the width of the groove and on the shape of the filling pieces; this angle is previously determined by careful calculation and experiment, and is decreased in the case of the final rings of blades near the exhaust end.

The necessary gradual increase in passage area between the blades is obtained either by increasing their length alone when the diameter of the rotor is constant (as fitted in the great majority of marine propelling turbines), or by increasing their length and the diameter of the rotor in steps (as commonly fitted in turbines used for electrical work). By either method it is possible to obtain the necessary increase of passage area with no alteration in the shape or spacing of the blades, which are therefore usually of a uniform standard section, except as regards length of projection and angle of setting.

In a few exceptional cases, where it has not been convenient to increase either the diameter of the rotor or the length of the blades, the necessary space for expansion has been obtained by blocking several of the passage-ways in the earlier and higher pressure stages,

and gradually increasing the number of openings towards the exhaust end of the rotor ; this is not a usual method, but a somewhat similar

Fig. 349.—Rotor-blading of Brush-Parsons Turbine.

idea is embodied in the Rateau and other turbines in combination with increased rotor diameters.

When the blades are long—above 2 inches, in ordinary cases—they are stayed together by (1) a circular wire ring passing through holes near their tips: or (2) a hoop of rectangular section (see Fig. 349) is partially embedded in the entrance (higher pressure) edges of the blades near their tips; or (3) the Willans and Robinson or other similar method of fitting semicircular rings of blades is adopted.

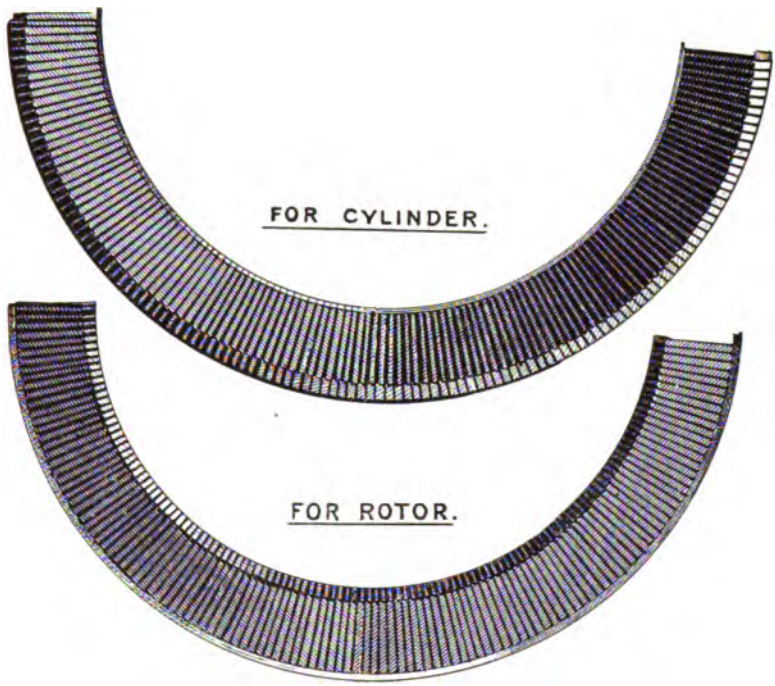


FIG. 350.—Half-Rings of Blades ready for fixing.

Method (2) is illustrated in Figs. 349 and 353. The ring, or hoop, and the blades are wired together, after carefully setting the tips of the blades at the correct distance from each other, and the ring, blades, and wire are soldered together with silver solder. For very long blades, two or three such rings are fitted and soldered. There is a tendency, when the blades are long and secured by an unbroken ring, for the blades to buckle from unequal expansion, and in the *Carmania* Messrs. John Brown & Co. introduced a sleeve connection in the ring or hoop in which one end of the binding ring can expand freely in a tube fitted on the opposite end.

In Fig. 350 two half-rings of blades ready for fixing, on the Willans and Robinson system, are shown: one for the cylinder, and one for the rotor. Referring also to Figs. 351 and 352, the tang or root of each blade is stamped mechanically into the form of an obtuse angle and dovetailed. The correct number of blades are assembled for a complete half-ring, by lightly riveting over a tongue left at the *tip* of each blade into the channel shrouding, and by pushing the wider part of the *tang* into one of the slots of the foundation ring. The riveting is done lightly, so as to leave the

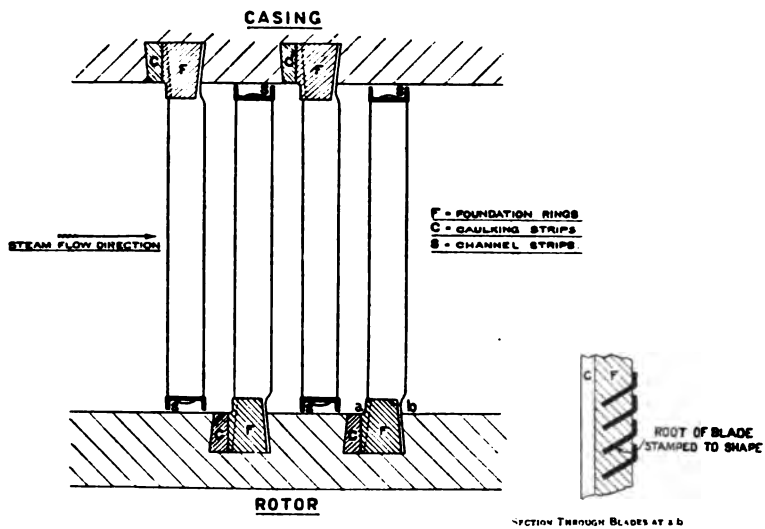


FIG. 351.—Section of Blading.

blades a certain freedom of movement in the shrouding, and thus enable them to be easily entered into the foundation ring. The half-ring, being so far completed, is next entered into the groove previously cut in the rotor (or fixed cylinder, as the case may be), and is fixed by means of the caulking strip *G*, which gives rigidity and strength. The dovetail form of the tang and caulking strip resist the centrifugal action, and is assisted in doing so by expanding the width of the caulking ring by very moderate caulking. After the ring is finally secured in the grooves, the outer ends, or tongues, of the blades are securely rivetted to the channel shrouding.

In the *Mauretania* the Wallsend Engineering Works machined the tip of each blade to a gouge point (see Fig. 349), and used the

ordinary Parsons system of a binding ring and wire lacing at the

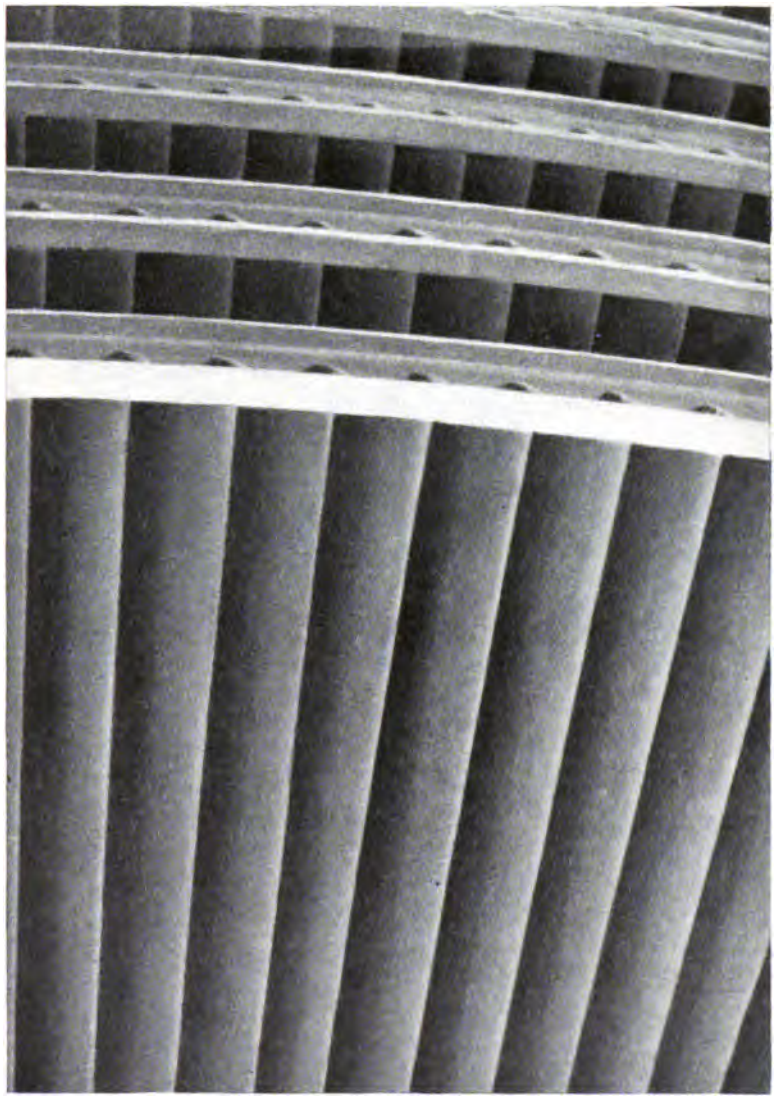


FIG. 352.—Blades and Channel-section Shrouding.

tips. *Engineering* states that "gouge pointing is now universally adopted to obviate any effect from the seizure of the blade against the

proximate surface when running." The same company has adopted the Parsons system of root-fixing for H.M.S. *Superb*, which is very nearly the same as that shown in Figs. 349 and 353, but a hole is drilled through the root of each blade and a corresponding hole through each filling piece so that they can be strung together on wire in segments and tightened up between cheeks. After completing the stringing, the blades are trued up in the vertical (radial) line, and the binding-ring and wire lacing soldered in place. The segments are caulked into the grooves, first with wooden mallets, and later with special caulking tools, so formed that they can be inserted between the blades to abut on the caulking (*filling*) piece—(*Engineering*, November 8, 1907).

The material of the blades is usually drawn section gun-metal

FIG. 353.—L.P. Rotor-blading.

wire; for superheated steam, nearly pure copper is preferable; and the best material, but very expensive, is made up of 80 per cent copper and 20 per cent nickel. In any case the surfaces of the blade must be perfectly smooth, and as frictionless as possible.

Height of Blades.—The makers and designers of Parsons marine propelling turbines have built up, as a result of experience, a number of formulæ which can be used successfully in turbine design. These formulæ are generally simple and largely empirical in character. The theory of the subject has been investigated very thoroughly, and theory is found to generally agree very well with practice. The empirical formulæ are usually based on a sound theoretical basis, although in some cases the foundation may appear somewhat obscure.

The first consideration in all cases is the rate of revolution of the turbine shaft (which is determined from the screw propellers)

thought to be most suitable from previous practice. Next comes the velocity of the steam, which depends on the upper and lower limits of pressure and the proportion U of the steam velocity which can be practically imparted to the blades while retaining a high efficiency. The steam velocity is divided into many components by the number N of rings of moving blades, and under normal conditions of about 150 lb. initial pressure and a high vacuum of about 28 inches in the condenser, the following formula is found fairly satisfactory—

$$U^2 N = \text{Constant} = 1{,}500{,}000;$$
$$\text{For the L.P. turbine, Constant} = 1{,}600{,}000.$$

U is generally considered to be measured at the surface of the rotor drum, but a measurement at the mean height of each ring of blades would, of course, be the mean blade speed.

The point of next importance is the height of the blades in the first ring, which practically governs the area of opening through the first ring of nozzles or fixed blades. This is determined from an estimate of the total weight of steam which must pass through the passages when the turbine is developing its full power. In this matter the allowance is generally much too liberal, and consequently turbines when completed are usually capable of developing more power than the boilers can supply. At present this is one of the worst features in the design of marine turbines. The diameter of the rotor is generally the largest which can be conveniently fitted within the space allowed, principally because a large diameter allows a shorter length of blade to be fitted at the exhaust end of the L.P. turbine. Considerations based on previous practice usually enable the designer to fix on the diameter of rotor which will best meet his requirements, and considerations of length of engine-room space generally determine the number of stages or separate turbines which should be employed.

With the *Mauretania* arrangement of one H.P. and one L.P. turbine, the drum diameters of 96 in. and 140 in. respectively agree with the practice of adopting a diameter ratio of 1 to $\sqrt{2}$. The following table (in conjunction with Fig. 346) shows some interesting details in regard to the actual blading, and to the pressures obtaining at the beginning of each expansion when the turbines are revolving at 194 revolutions per minute (= 26·44 knots) :—

	H.P. Turbine.			L.P. Turbine.		
	Gauge Pressure.	Height of Blades.	No. of Rows.	Gauge Pressure.	Height of Blades.	No. of Rows.
	lb.	in.			in.	
Receiver	150	2½	15	5 lb.	8	7
2nd expansion	113	3⅛	15	Atmosphere	10½	7
3rd ,,	87·5	4	15	4 in.	13½	7
4th ,,	63	5	15	9¾ ,,	18	7
5th ,,	43	6¼	15	16½ ,,	22	8
6th ,,	31½	7½	15	21½ ,,	22	8
7th ,,	21½	9⅜	15	24 ,,	22	8
8th ,,	13	12	15	26 ,,	22	8

Exact heights uncertain at time of writing.

Uncertain.

Boiler pressure, 180 lb.; condenser, 28 in.; barometer, 30·1 in. The blading allows only about one-third of the annulus as a passage-way for the steam at the inlet end of the H.P. turbine. A passage-way of 86 per cent of the annulus at the condenser end of the L.P. turbine is obtained by the adoption of "wing" blades which are nearly parallel to the rotor axis.

At the designed speed of 190 R.P.M. the surface of the H.P. rotor has a velocity of nearly 80 feet per second. At the same time the L.P. drum has a velocity of about 116 feet per second.

Using the formula quoted above, and considering the H.P. turbine velocity only—

$$\text{The number of rotor rings} = N = \frac{1{,}500{,}000}{80^2} = 234 \text{ about.}$$

But as only one-half the power is developed in the H.P. turbine, the number of rings is only one-half, viz. 117, which is somewhat less than the actual number fitted (120 about).

Considering the L.P. turbine (for which a constant of 1,600,000 may be used)—

$$N = \frac{1{,}600{,}000}{116^2} = 119 \text{ about.}$$

As before, only one-half the power being developed in the L.P. turbine, the number of rings $= \frac{119}{2} = 59{\cdot}5$, which is very nearly the actual number fitted (60 about).

From certain illustrations of these turbines, it appears that the H.P. rotor blading is arranged in 8 expansions with 15 rows of blades in each. The height of the intermediate rows has not been

published, but the first or high pressure row is $2\frac{1}{2}$ inches and the last row 12 inches in height. The L.P. rotor blading appears to be arranged in 5 expansions, but owing to the "winging" of the blades in the rows near the exhaust end there are probably 8 expansions, with 7 rows in each of the first four and 8 rows in each of the last 4 expansions. The blades at the inlet end are 8 inches and those in the last 4 expansions are 22 inches in height, but as in the case of the H.P. turbine the exact figures are not available for the intermediate rows.

For turbines constructed previously to those of the *Mauretania*, the blade heights of each expansion were increased in geometrical proportion, the common ratio being $\sqrt{2}$, or about 1·4. For the *Mauretania*, however, a lower common ratio appears to have been successfully adopted, which appears to be about 1·25 for the H.P. turbine and 1·31 for the L.P. turbine.

The remarks given above may perhaps assist in solving a few of the early troubles experienced in designing a pressure turbine, but it is beyond the scope and intention of this work to penetrate further into the subject.

Balancing Axial Thrust.—In all axial-flow turbines which embody reaction, such as the Parsons type particularly, there is an axial thrust on the shaft in the same direction as the flow of the steam. This thrust is due primarily to statical differences of pressure on the moving blades between the inlet and exhaust. Movement of the rotor and blades, within the fixed cylinder, in an axial direction is prevented by a thrust block and collars on the shaft, which are described later. No work is done when no motion is produced, and, consequently, the axial thrust does not occasion expenditure of energy except in overcoming the friction in the thrust block caused by the difference of pressure. In Parsons turbines this statical thrust is balanced by *dummy pistons* fitted on, and incorporated with, the rotor drum, so that as little pressure is exerted on the thrust collars as possible, and thus thrust friction is almost entirely eliminated.

A common method of balancing the axial thrust of turbines, other than those used for marine propulsion, is shown in Fig. 354 (Westinghouse-Parsons type). The steam enters at *A*. Blading is fitted in connection with the various parts, *B*, *C*, *K*, *K*, *L*, *M*, *D*, of the rotor, and the difference of inlet and exhaust pressure tends to push the rotor to the right, or exhaust, end. Dummy pistons, *E*, *F*,

and G, are fitted to the left of the inlet A, and by suitable channels in the casing the difference of pressure on these dummies tends to push the rotor to the left and thus balance the axial thrust on the blades. The diameters and areas of these dummy pistons are made suitable to the various total pressures corresponding to the various

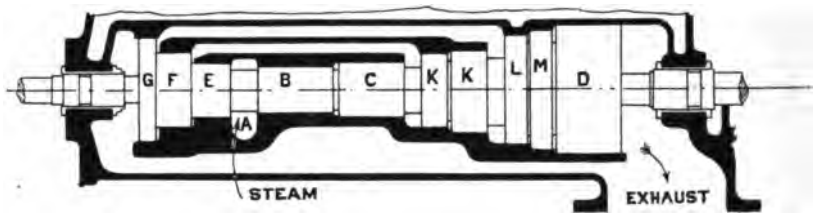


FIG. 354.—Balancing Axial Thrust.

overall diameters of the successive enlargements of the rotor and increased length of blades.

The Willans and Robinson method of balancing axial thrust in an electric generating turbine engine is shown in Fig. 355. A, A', A'' is the annular channel in which the blades (fixed and moving) are fitted. BB is the exhaust. C is an annular face exposed to

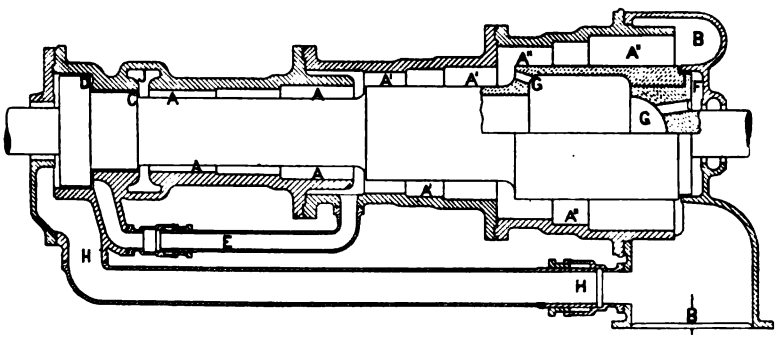


FIG. 355.—Balance Pistons of Willans and Robinson (Parsons type) Turbine.

high pressure, which is admitted at J, and the pressure on C balances the thrust pressure on the moving blades in the space AA. D is an annular face exposed to the steam pressure between A and A' through the connection E, and the pressure on D balances the thrust pressure on the moving blades within the space $A'A'$. The thrust pressure on the low pressure and longest blades, $A''A''$, is balanced by admitting pressure (about atmospheric) through holes, G, to the interior of the rotor and the end F of the turbine; this

pressure being greater than that of the exhaust at B, and consequently, through the passage H, of the pressure behind D (to the left), exerts a thrust from right to left and balances the axial thrust on the moving blades $A''A''$ from left to right. In connection with this system it should be noticed that the shaft packing gland at F is exposed to approximately atmospheric pressure, and at the opposite end to the exhaust pressure, which in a condensing engine is considerably less than atmospheric, and consequently there is little trouble in obtaining steam and air-tight glands.

Many systems have been invented for entirely eliminating the dummy pistons and balancing the axial thrust. Davey's, Carnt's, and others are examples of these systems, but up to time of going to press nothing is known of their practical application or efficiency.

Reversing.—A little study of the form of the Parsons blading (Fig. 348) makes it evident that the turbine is not reversible by changing over the steam and exhaust, and consequently it is necessary to use a separate turbine for reversing the direction of motion of the shaft, unless gearing (which would be impossible for any but the lowest powers) is used for the purpose. For marine propulsion, the power and speed required for astern working can be less than that used for going ahead, and consequently the astern turbine is smaller, and also less economical, than the ahead turbine. In many cases it has been found convenient to install the astern turbine within the exhaust casing (as shown in Fig. 356) of the L.P. turbine. In this position the astern turbine L revolves with the ahead turbine K and in the vacuum created by the condenser and air pump; in consequence of this low pressure the astern turbine offers a very low resistance when revolved by the working of the ahead turbine. In a similar manner, when the astern turbine is at work, the ahead turbine revolves in a medium of low resistance, viz. the vacuum created by the condenser and air pump. A valve and pipe are fitted for connecting the steam ends of the astern and ahead turbines respectively for ahead and astern working, with the exhaust space.

Referring to Fig 356, K is the L.P. ahead turbine of a series of turbines, and steam enters at A from another turbine. After passing through successive rings of fixed and moving blades, B, C, D, E, etc., the steam enters the condenser through the exhaust space H. For entering and leaving harbour, etc., the steam inlet

A is closed by a self-closing valve, and high-pressure steam can be admitted directly from the boiler either through the pipe *P for ahead working* or through *O for astern working*. The blades of the astern turbine are fitted with reference to the axis in an opposite manner to that in which the ahead blading is fitted, so as to produce motion in the opposite direction.

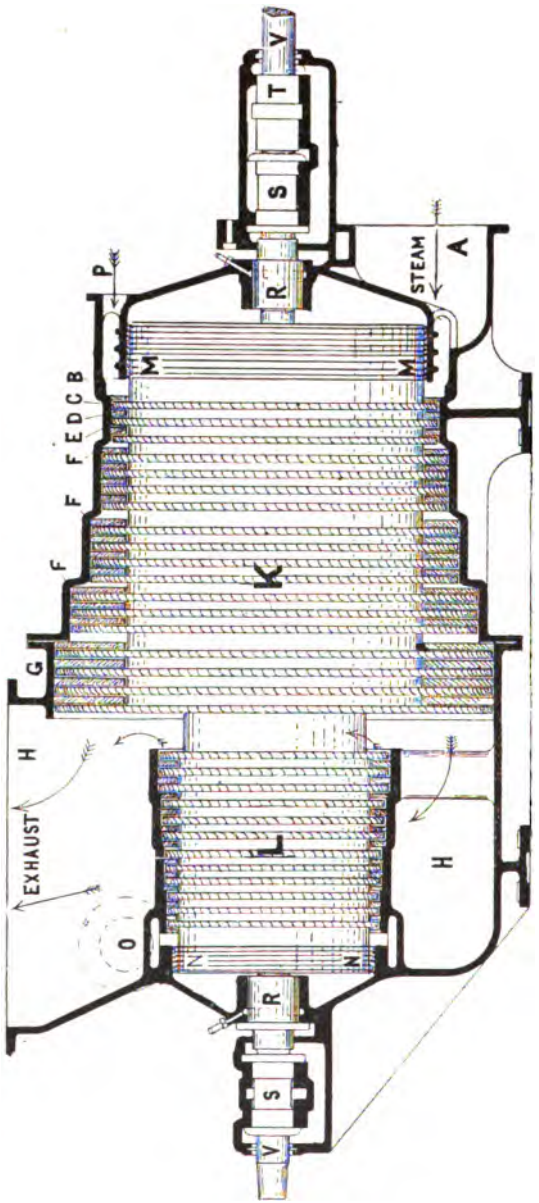


FIG. 356.—Astern Turbine and L.P. Ahead Turbine.

The moving blades of both ahead and astern turbines are mounted on the hollow drum *LK*, which is carried by shafts, *V*, and bearings, *S*, near the ends of the fixed casing. Steam packed glands are fitted at *R*, and a thrust block, usually at the forward end of the ahead turbine, at *T*. To prevent the passage of high-pressure steam into the exhaust space, dummy packing rings are fitted at *M* and *N*, and by this means axial balancing is partly effected.

In some instances, now becoming usual with large powers, the

astern turbine is an entirely separate fitting to the ahead turbine, but is fitted on the same shaft, as shown in Figs. 369 and 370.

Dummy Piston Packing.—The dummy piston, in addition to absorbing part of the axial thrust, is used to prevent the passage of steam from the inlet to the exhaust without acting on the blades. Grooves, as shown in Fig. 357, are cut in the outer circumference of the dummy (or part of the rotor equivalent to it) at *M* (Fig. 356). These grooves are made with a slightly projecting lip, and a corresponding number of inwardly projecting rings are fitted in the fixed casing or cylinder so as to enter the spaces formed by the grooves. By careful adjustment of the axial position of the rotor within the fixed casing, using the thrust block for this purpose, the fixed rings barely touch the lips of the grooves, and leave a very small space

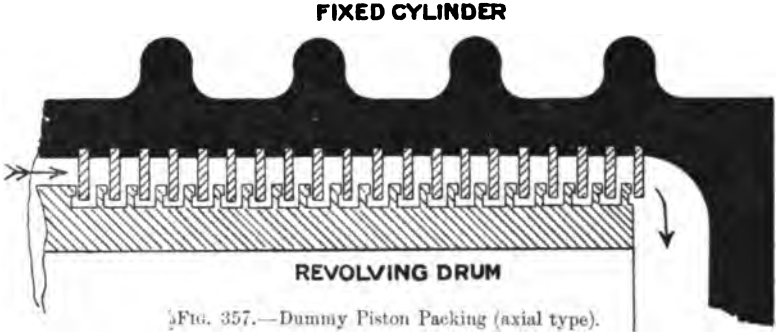


FIG. 357.—Dummy Piston Packing (axial type).

through which the steam can leak. The centrifugal action of the rotor collars between the grooves assists in preventing leakage, and the escaping steam expanding into each labyrinth space in turn through a number of such expansions, twenty or more in practice, the total leakage is thus extremely small. The frictional resistance of the dummy packing to the rotation of the drum is almost inappreciable, because there is no actual metallic contact of the lips and rings. For the same reason internal lubrication is unnecessary. The method shown in Fig. 357 is generally adopted for the ahead inlet end of the turbine and nearest the thrust block.

For the astern turbine end, allowance has to be made for difference in expansion of the material of the casing and rotor, and the axial type of labyrinth packing is not susceptible of sufficiently close adjustment. A radial type of packing is therefore generally adopted, as shown in Fig. 358, which, however,

embodies the same general principle. Rings of thin gun metal (now usually V-shaped at the proximate edge) are fitted so as to project outwardly from the rotor and inwardly from the casing, so that they nearly touch the opposite fixed or moving parts respectively. Plenty of room is allowed axially between each pair of fixed and moving rings, so that there is no possibility of metallic contact.

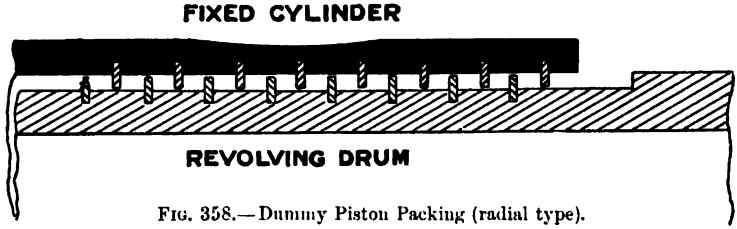


FIG. 358.—Dummy Piston Packing (radial type).

It has already been stated that movement lengthways is prevented at the forward end by the thrust block, and therefore expansion must be allowed for in the turbine casing towards the after end. This is obtained by elongating the holes in the casing standards, in an axial direction, for the holding-down bolts. The axial clearance between the fixed and moving blades is much larger than the clearance between the dummy piston rings, and thus prevents the blades from coming in contact with each other.

Shaft Packing Glands.—The glands *R*, in Figs. 356 and 360, at the parts of the shaft where it leaves the casing, are steam packed. In Fig. 360 a number of split Ramsbottom rings fit loosely into grooves between collars on the shaft inside the gland, and are free to move along the casing and to expand, and they are usually allowed to revolve with the shaft. This gland is not used in any way as a bearing, and only serves to prevent the admission of air into the exhaust space by admitting a very slight quantity of steam or water, which does not reduce the vacuum. The steam or water, either low-pressure steam or hot-well water, is admitted through a small pipe in connection with the grooves, and situated about one-third the length of the gland from the outer end. In cases where the internal exhaust pressure is greater than atmospheric pressure, such as the H.P. turbine, the steam pipe is connected through a two-way cock with the condenser. An alteration in power, producing a variation in the exhaust pressure, necessitates a slight readjustment of the steam pressure in the

packing gland; a slight external leakage, which causes no inconvenience, is generally allowed at all powers. No oil is admitted to these glands for lubrication, and as there are no internal bearings or rubbing surfaces in the turbine, internal lubrication is unnecessary.

A ring, *W*, is shrunk on to the shaft, to throw off any oil that may work along the shaft from the bearing *S*, and it thus prevents oil working into the packing gland and exhaust steam.

In some of the more recently constructed vessels a modified type of spindle gland packing is used, as shown in Fig. 359. The part next the exhaust space is fitted with V-shaped rings on the

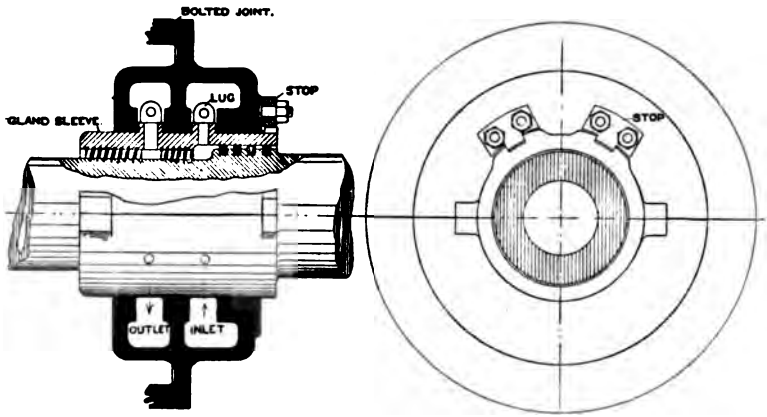


FIG. 359.—Shaft Packing Gland.

labyrinth principle, used for the dummy radial packing, and the outer part is fitted on the Ramsbottom ring principle described above. Steam of moderate pressure is admitted to the inlet space and enters the space round the shaft through holes drilled in the gland sleeve, and finds its way out through the outlet at a reduced pressure through holes similarly drilled in the sleeve between the inner and middle sets of packing. The gland sleeve is contained within a casing secured by a bolted joint to the turbine casing. The sleeve casing is made in two semicircular parts, and when disconnected the two parts can be moved clear of the sleeve. The sleeve is also made in two semicircular parts and can be removed from the spindle for examination of the packing. The whole arrangement as described thus allows the gland packing to be examined or renewed without lifting the upper half of the turbine

casing. A slight axial movement of about $\frac{20}{1000}$ths of an inch is allowed for the gland sleeve, which is prevented from revolving by the stops shown in the figure.

Shaft Bearings.—Only two bearings (*S* in Figs. 356 and 360) are necessary for the turbine itself, one at each end of the case. The bearings are of white metal, contained in cast steel or brass, and are supported on adjustable pieces fitting into the engine framing. The white metal fits the shaft accurately over nearly the whole bearing area; there is little clearance at the butts compared with a reciprocating engine bearing. Oil is *forced* into the

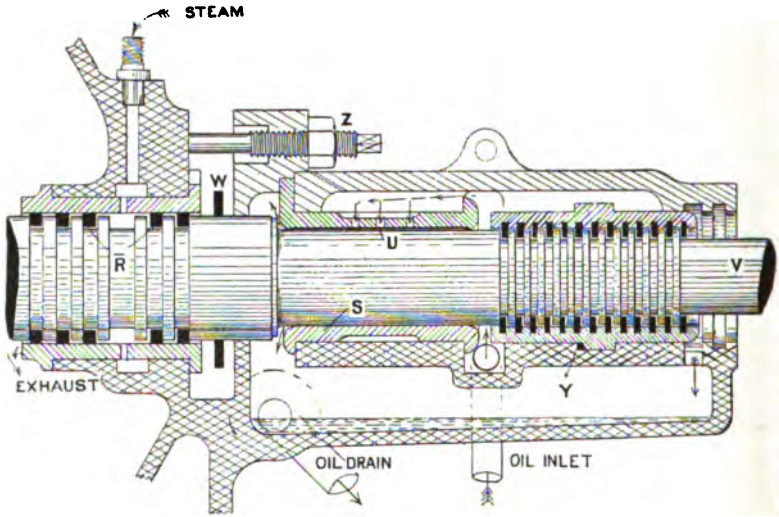


FIG. 360.—Shaft Bearing and Thrust Block.

top half of the bearing by a pump, through holes *U* in Fig. 360, near its mid-length, and works out at the ends into an oil well; an oil groove is cut along the top of the bearing, leading from the inlet holes to nearly the ends of the white-metal lining. The oil gravitates through a drain pipe into a tank fitted with a cooling coil, and, after being filtered, is returned back through the bearings by the pump. The thrust bearing is lubricated in the same way, which, as the bearings run for some years without any appreciable wear, is most successful.

For small engines, the main bearings are made in one piece and slipped over the end of the shaft, with three concentric steel rings or cylinders round each. There is about $\frac{1}{1000}$ part of an inch play

between these cylinders, and the spaces, filled with compressed oil, allow the shaft to be more rigidly held, without becoming overheated, and allay some tendency to wobbling of the shaft in its bearings.

Another type of bearing, which is used to prevent wobbling, or chattering, is made by cutting one of the concentric rings into three segments; this ring is left rather smaller in diameter than the cylinder it supports, and consequently each segment acting as a coach spring tends to prevent the chattering of the shaft. The use of forced lubrication assists the bearings in steadying the shaft, by imposing a nearly incompressible but still slightly elastic buffer between the concentric parts of the bearing.

Axial Thrust, Bearings, and Adjustment.—It has already been stated that the difference of pressure of the steam promotes an end thrust along the axis of the shaft. When a marine screw propelling turbine is turning ahead, this end thrust is nearly balanced by the thrust of the propeller; assuming, of course, that the steam travels along the drum from forward to aft. Part of the difference is balanced by a *dummy piston* (*M* in Fig. 356) formed at the forward end of the drum. Any other slight difference is absorbed by an external thrust bearing.

The *thrust bearing*, *T* in Fig. 356, is usually fitted at the forward end of the main propelling turbine, just clear of the casing. The bearing is made in two halves, upper and lower, the former of which is adjustable along the shaft by studs, *Z* in Fig. 360, with check nuts and butting on the casing. The collars on the shaft bear on separate half-rings, which are fixed in the upper and lower parts of the bearing, and fit loosely between the collars.

The *end play* of the drum is adjusted by drawing the shaft forward until the dummy piston (*M* in Fig. 356) bears hard up against its collars; then a ring is fitted at the after end *Y* (Fig. 360) of the collar on the lower part of the thrust block, of sufficient thickness to set back the dummy the required amount of clearance. The end clearance of the dummy packing rings varies from 0·003 to 0·025 of an inch, according to the type and size of turbine. The upper half of the thrust block is then adjusted, to prevent the shaft working aft, and to take any difference of end thrust, when the rotation of the propeller is reversed for going astern.

Vibration Balancing.—The rotating parts of a turbine tend to set up vibration, if they are not symmetrically arranged about the

axis of rotation. Great care is taken in the manufacture by machining the drums both inside and out, as well as the shaft and other parts, but notwithstanding this precautionary measure there is still some tendency to cause vibration. After the rotor is completed and connected with its shaft, it is usually mounted on two perfectly horizontal knife-edged supports under the bearing positions. Any bias tending to retain the rotor in any particular vertical plane is corrected by removing or adding weight to compensate the bias. The above is termed the *static balancing.*

It is possible that static balancing may not suffice to stop vibration when the rotor is revolved at a high speed, and in many cases *dynamic balancing* is necessary. For this purpose the turbine is put together, and the rotor revolved at its highest speed, when tested for vibration by means of standard fittings or a vibrometer. The corrections are made by a series of trials until vibration is practically eliminated; weight being added or removed as thought necessary at certain parts. The process is almost entirely experimental and may frequently be extremely tedious, but good balancing always conduces to better working quite apart from the inconvenience caused by vibration.

Relief Valves.—Above the exhaust casing of each turbine a large pipe is fitted with its upper end above the upper deck, and a relief valve, or safety valve, is fitted at some convenient part of the pipe *outside* the engine room. The fracture of the shaft, or other accident, might allow the turbine to revolve without doing any external work, and thus allow too great a pressure to accumulate in the exhaust casing and condenser; the relief valve prevents such accumulation of pressure.

Steam Jacketing and Superheating.—The expansion is continuous, the fall of pressure and temperature is also continuous, and consequently, when the engine is working steadily, the temperature is very nearly constant at any particular point in the cylinder. Cylinder condensation is therefore practically non-existent, and the use of jacketing, although it has been tried, does not appear to justify the extra weight and complication involved.

On the other hand, superheating is efficient, and the economy obtained by its use is about 8 to 12 per cent. The superheat given to the steam counteracts the condensation caused by the conversions of heat into kinetic energy and mechanical work; any superheat added to the quantity necessary to effect this purpose is not likely to

further improve the efficiency. The limiting temperature of efficient superheating for turbines is probably about 120° to 140° Fahr. above that of the steam generated in the boiler.

It is stated that, when the steam supplied to the turbine is much superheated, the radial clearance space must be much increased to allow for the consequent, but uncertain amount of, expansion of the blades at the higher temperature. The perfect regulation of the exact rise in temperature of superheat has not yet been obtained in practice. When such regulation is possible, the now necessary increased clearance of the blades may not be required. It is quite possible that the necessary increase of radial clearance, for using superheated steam, may cause a loss which nearly balances the increased economy obtainable by superheating.

In the turbine the presence of water in the steam increases the frictional resistances of the blades and retards the velocity of passage, and thus reduces the mechanical efficiency. The presence of *priming water* is not, however, dangerous, as it is in the reciprocating engine; the turbine is reduced in speed and efficiency, but no damage is done.

Steam Consumption.—The efficiency and economy of the marine turbine have been greatly increased in the last few years, and it now compares favourably with the reciprocating engine of the best design at full power; but the economy of the turbine is at very low powers less than that of the reciprocating engine.

Turbines of great size and power give relatively greater economy than small ones, and this is in favour of their adoption for large vessels. The greater diameter of the drum (on which the blades are fixed) produces a greater and more efficient leverage, while the frictional resistance is very slightly increased compared with the greatly increased power obtainable. Clearance at the tips of the blades, however, must probably be increased with the size of turbine, and there is thus some tendency to increase the loss through the clearance spaces.

Turbines have their full measure of economy from one-half to full power, and at one-fourth the economy is good; but at one-sixth to one-twentieth, which is the cruising power for high-powered warships, the economy is not good, and special means are now used to improve the economy at low powers.

Arrangements for obtaining Economy at Low Powers.—The ratio of expansion in an ordinary reciprocating engine is the ratio

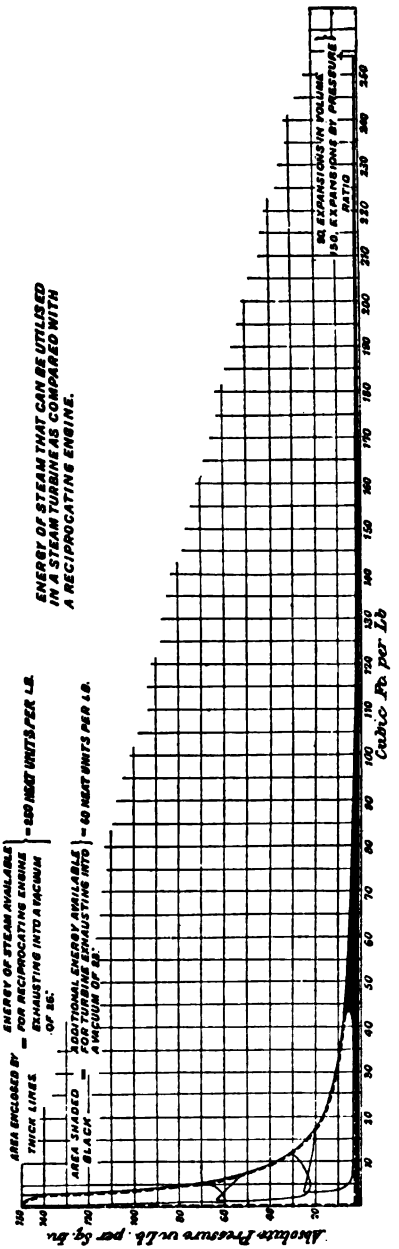

ENERGY OF STEAM THAT CAN BE UTILISED IN A STEAM TURBINE AS COMPARED WITH A RECIPROCATING ENGINE.
AREA ENCLOSED BY THICK LINES = ENERGY OF STEAM AVAILABLE FOR RECIPROCATING ENGINE EXHAUSTING INTO A VACUUM OF 28". = 250 HEAT UNITS PER LB.
AREA SHADED BLACK = ADDITIONAL ENERGY AVAILABLE FOR TURBINE EXHAUSTING INTO A VACUUM OF 28". = 40 HEAT UNITS PER LB.
Absolute Pressure in Lbs. per Sq. In.
Cubic Ft. per Lb.

FIG. 361.—Energy of Steam.

of the volume occupied by the steam in the L.P. cylinder at the instant of release, to that occupied by the steam in the H.P. cylinder at the instant of cut-off. With a ratio of L.P. cylinder to H.P. cylinder o about 7 to 1, and a cut-off in the latter at about half-stroke, the ratio of expansion is about 14, neglecting clearance; by linking up for low powers, the ratio is increased to about 20 to 24.

In the turbine the steam is expanded down to almost the exhaust pressure, as shown in Fig 361,[1] and the ratio of expansion is practically governed by the ratio of the steam pressure on admission to the pressure of exhaust. Assuming that the admission pressure is 150 lb. absolute, and that the exhaust pressure is 2 lb. absolute, the ratio of expansion becomes about $\frac{150}{2} = 75$. The final, or condenser, pressure is practically constant, and no increase of expansion is possible except by increasing the admission pressure.

At low powers the pressure of admission is reduced by throttling or wiredrawing, and the ratio of expansion is *reduced* in the turbine; but in the reciprocating engine, as we have

[1] From *Engineering*, October 5, 1906. Paper by Hon. C. A. Parsons and R. J. Walker.

seen, the ratio of expansion may be *increased* by linking-up. Consequently the economy of the turbine at low powers cannot be so good as that of the reciprocating engine, unless some special method is devised for utilising a higher admission pressure and a high ratio of expansion. Several plans have been tried (and are under construction) to obtain a better economy for ships at low powers—one by using a reciprocating engine in conjunction with the turbine; and another by using a smaller turbine, called the cruising turbine, in conjunction with the main turbine.

The number of fixed blades or distributers in "impulse" turbines is sometimes limited in the fixed rings, and the usual passage-ways at certain parts are left solid. The number of passage-ways is gradually increased in the succeeding rings, to allow expansion of the steam; the length of blades, or the diameter of the rings, does not usually increase in this type. The moving blades and spaces are complete, and fitted in the usual way.

SS. "Carmania."—The general arrangement of the turbines is shown in Fig. 362, and this vessel makes a distinct link in the chain of adoption of turbine engines from small to large vessels. The *Carmania* displaces 30,000 tons when loaded, and the turbines develop about 20,000 shaft H.P. when running at 180 revolutions per minute for the full speed. The propellers are three-bladed, and one of 14 feet diameter is fitted on each of the three shafts.

The H.P. turbine rotor is about 8 feet in diameter, and is fitted in a casing on a somewhat lower platform than the two L.P. turbines, which are fitted on the wing shafts. The L.P. turbine rotors are 11 feet in diameter by $8\frac{1}{2}$ feet in length, and are made of forged steel $2\frac{1}{2}$ inches in thickness. The blades are 2 inches long at the inlet end and increase in steps to $7\frac{1}{2}$ inches at the condenser end. The total number of turbine blades is about 1,150,000. Including the reversing turbines fitted inside the L.P. exhaust casings, the overall length is about 36 feet from centre to centre of the rotor shaft bearings.

The run of the steam piping and the position of the various valves is shown clearly in Fig. 363. The manœuvring valves are fitted at *M*, and each side can be shut off when not required by a stop valve, *S*; this valve is necessary, because the manœuvring valves cannot be made very close fitting. Self-closing valves, *F*, are fitted in the L.P. receivers, so that when the manœuvring valves are in use steam pressure cannot reach the H.P. turbine.

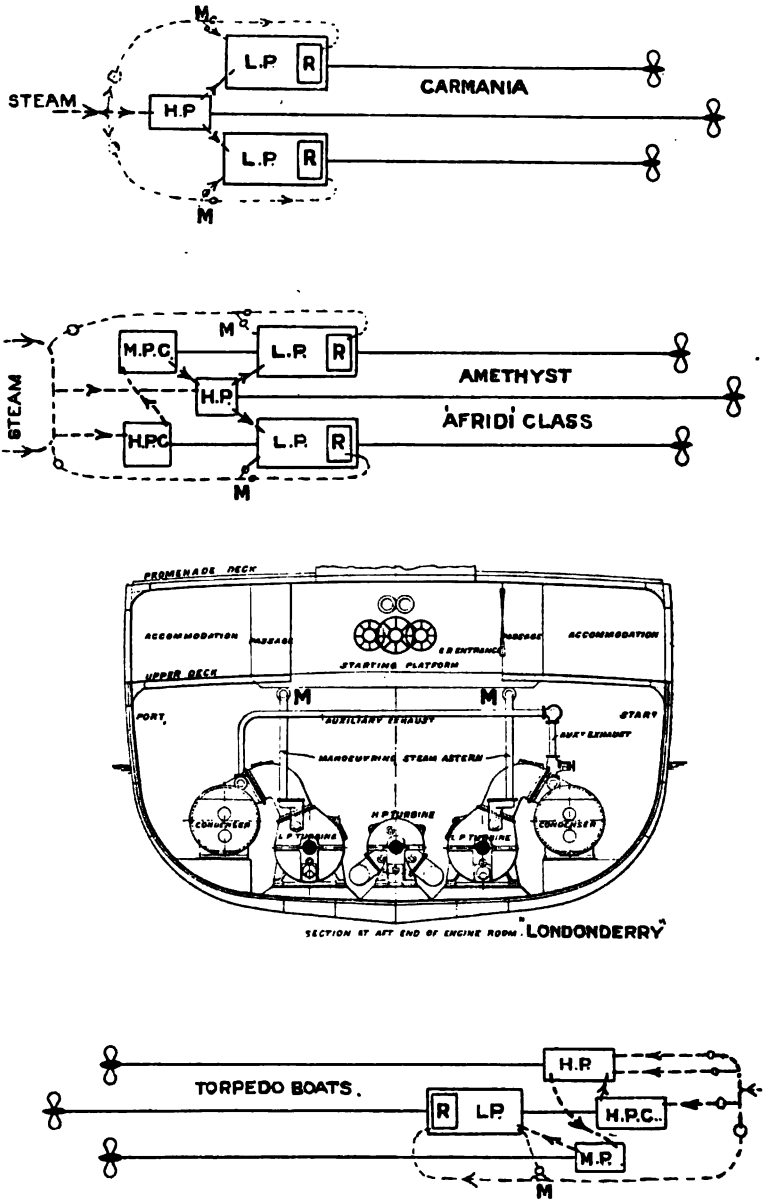


FIG. 362.—Three-shaft Arrangement of Marine Turbines.

Arrangements of Marine Turbines.—Fig. 362 shows various arrangements of turbines for driving three propeller shafts as fitted in the vessels named. In all cases it should be remembered that an *idle* turbine (*i.e.* one not actually steam-driven) is free to revolve with its screw propeller, and both steam and exhaust ends of an idle turbine are connected temporarily with the exhaust, so that the rotor blades move *in vacuo*, *i.e.* in a medium offering very low frictional resistance. The dotted lines show the general run of the steam piping connections.

SS. *Carmania*—

Three shafts; three screw propellers. One H.P. and two L.P. ahead turbines; with two-stage expansion. Each L.P. turbine can be worked independently for manœuvring. Manœuvring valves and gearing are fitted at *MM* for this purpose, which also control the two reversing turbines *R* (inside L.P. exhaust casings), one on each wing shaft.

H.M.S. *Afridi* (torpedo-boat destroyer) and *Amethyst*—

Three shafts; three screw propellers.

Main turbines: one H.P. and two L.P. ahead turbines; two-stage expansion for high powers. Each L.P. turbine and two reversing turbines can be worked independently for manœuvring.

Cruising turbines: one H.P.C. and one M.P.C. ahead turbines; four-stage expansion for low powers, or, with H.P.C. shut off, three-stage expansion for moderate powers.

SS. *Londonderry*—

Section at aft end of engine room showing general arrangement of three shafts and turbines with two condensers.

H.M. *Torpedo Boats*, Nos. 1 to 12—

Three shafts; three screw propellers.

Main turbines: one H.P., one M.P., and one L.P.; three-stage expansion for high powers.

Cruising turbine: one H.P.C. on middle shaft; four-stage expansion for low powers.

One reversing turbine on middle shaft only.

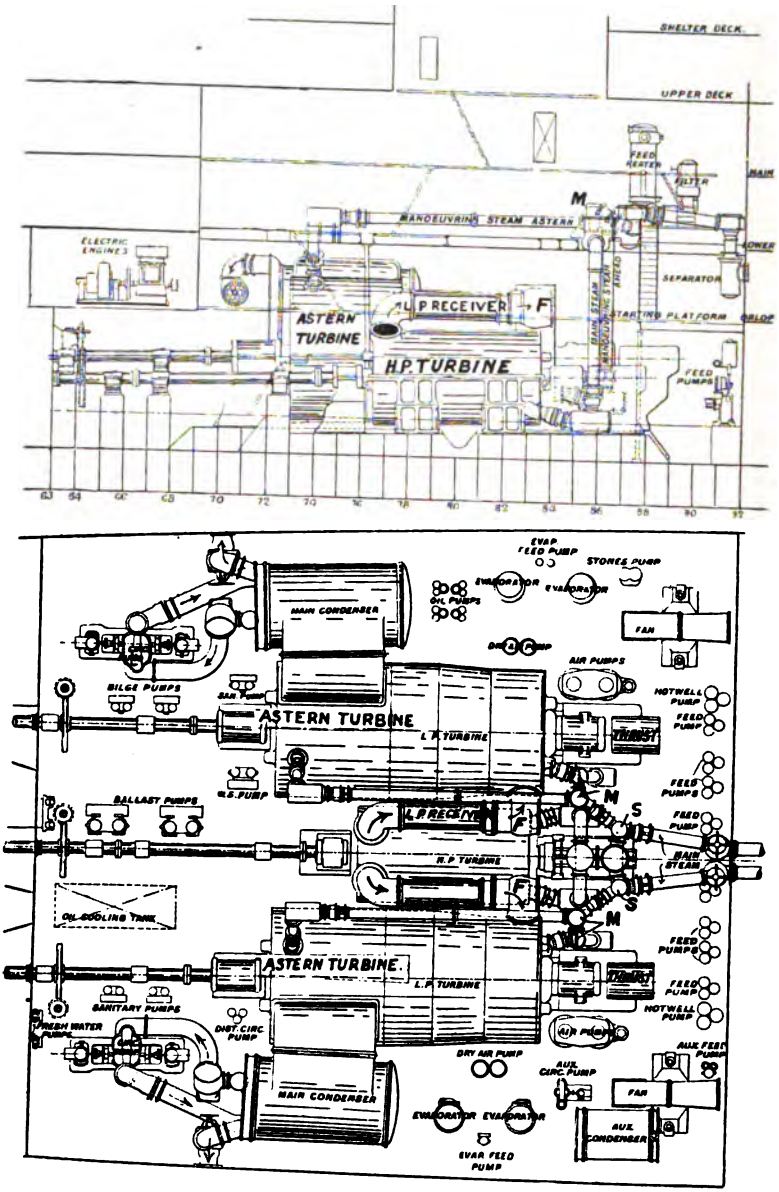


FIG. 363.—Turbine Engines of ss. *Carmania*.

The *Carmania* is a sister vessel to the *Caronia*, the general arrangement of whose engines is shown in Fig. 132.

In several torpedo-boat destroyers and the third-class cruiser *Amethyst* the three-shaft arrangement consists of one H.P. turbine on the middle shaft and one L.P. turbine on each of the two outer shafts. Inside the exhaust casing of each L.P. turbine an astern turbine is fitted; and two small cruising turbines are fitted, one on

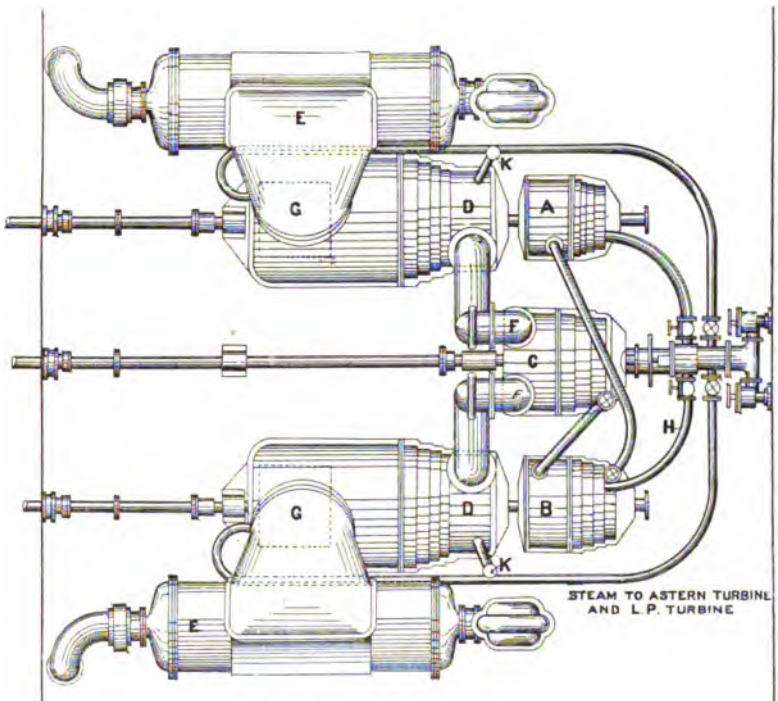


FIG. 364.—*Amethyst* Turbines.

each of the outer shafts, in front of the main propelling turbines. The arrangement is shown in Fig. 364.

For the lowest powers and speeds, steam is admitted to the H.P. *cruising* turbine *A*, and the exhaust steam from it is led into the other, the M.P. *cruising* turbine *B*, where it does more work before being exhausted into the *main* H.P. turbine *C*, on the middle shaft; thence the steam passes into the two *main* L.P. turbines *DD*, and finally escapes into the condensers *E*. The steam is thus used in four stages, with a consequent reduction of velocity in each

stage, and as steam of a higher admission pressure can be used in this way, the rate of expansion and economy are greater than when using only two stages.

For powers which cannot be quite obtained in the above manner, a little extra steam is admitted directly, through the valve and pipe, *H*, into the M.P. *cruising* turbine, which is the second stage of the series.

For still higher powers, steam can be admitted directly into the *main* H.P. turbine, and, if necessary, the steam is shut off from the two cruising turbines, which then revolve idly with the main L.P. turbines.

For temporary increases of power, beyond the designed power, steam can be admitted to the L.P. turbines directly; but this additional supply must be made with discretion, and as a rule is used for manœuvring only. Self-closing valves, *F*, are fitted to prevent any back pressure returning to the main H.P. turbine, when steam is admitted directly to the main L.P. turbine or turbines. At *K*, the manœuvring valves are fitted for controlling the steam for working the wing shafts ahead or astern.

The *Amethyst* was the first cruiser to be fitted with turbines, and the Admiralty allotted the same machinery weights to each of four sister vessels, including the *Amethyst*. The comparative results are shown in the following table of steaming distances:—

Speed.	*Amethyst* (Turbines).	*Topaze* (Reciprocating Engines).
At 10 knots	5570 miles	7300 miles
14 ,,	4950 ,,	5100 ,,
18 ,,	3600 ,,	2770 ,,
20 ,,	3160 ,,	2140 ,,
22 ,,	...	1420 ,, (full speed)
23½ ,,	1620 ,, (full speed)	...

The advantages at high speeds of the turbines are thus clearly shown; but notwithstanding the cruising turbines, which are used at speeds up to 20 knots, the *Topaze* reciprocating engines had some advantage at speeds below 14¼ knots, at which speed about one-fifth the full power is developed. Subsequent to these trials, the exhaust steam from the auxiliary engines has been conducted into the L.P. ahead turbine receiver pipes, and by this

means the steaming distance of the *Amethyst* has been increased, and is now believed to compare favourably with the *Topaze* and her sister vessels down to 10 knots, as shown in Fig. 364A. (R. J. Walker, M.I.N.A., Liverpool Engineering Society, February 7, 1906.)

H.M.S. *Mohawk*, the general arrangement of whose engine rooms is shown in Figs. 365 and 366, may be considered as typical

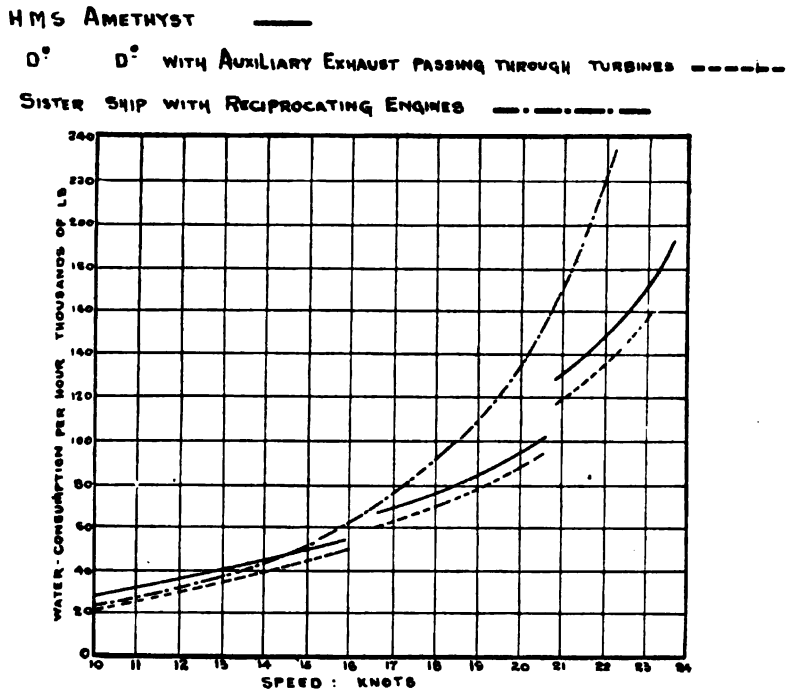


FIG. 364A.—Feed-water Consumption of H.M.S. *Amethyst* and *Topaze*.

of recent three-shaft arrangements in warships, and is similar in character to the outline sketch shown in Fig. 362 for the *Afridi*. The figures illustrate the position of the various valves and pipe connections necessary for working the turbines from the starting platform, which is at the mid part of the engine room between the L.P. turbines and just abaft the H.P. turbine. Fig. 365 shows the position of the various operating wheels and gearings, and Fig. 366 shows the various valves and pipes in connection with them. For each of the seven separate turbines, there is a valve and steam

connecting pipe leading direct from the boiler steam pipe. There are two main steam pipes leading from the boilers to the engine-room bulkhead, and the supply of steam to the turbines is controlled by two bulkhead valves, *A*, with a cross connecting pipe, *C*.

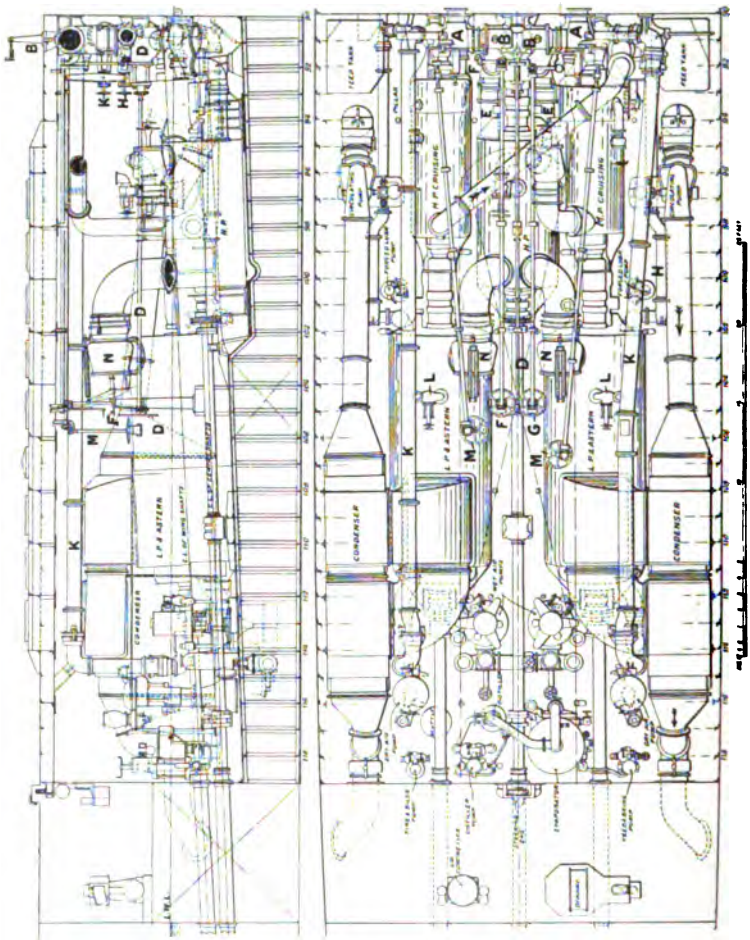


Fig. 365.—General Arrangement of Turbines in *Mohawk*.

In this cross connecting pipe there are also two emergency shut-off valves, *B*, which are workable from the upper deck. One valve, *D*, controls the supply to the H.P. turbine through two pipes, *E*. Direct supply to the H.P. cruising turbine is controlled by the valve *F*, and to the M.P. cruising turbine by the valve *G*. Direct

supply for ahead and astern working of the turbines within the L.P. casing is controlled by two manœuvring valves, *H* and *K*, on each side, worked by a set of gearing, *M*, so that when one valve is

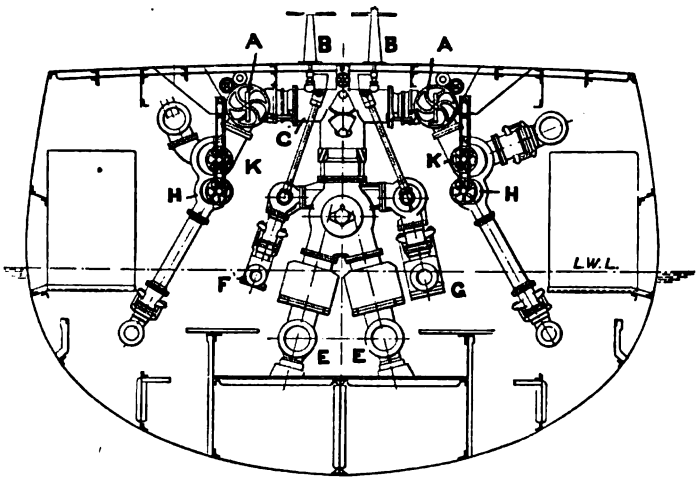


FIG. 366.—Steam and Manœuvring Valves in *Mohawk*.

open the other remains closed. Connections are made through *L* for the closed (auxiliary) exhaust system. Self-closing valves are fitted at *N*, the compensating springs for which are shown in

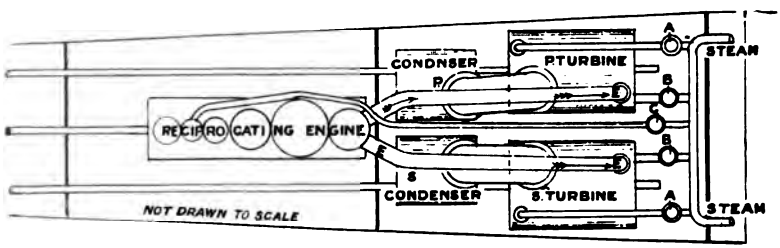


FIG. 367.—Combined Reciprocating and Turbine Engines.

Fig. 365. The cruising turbine rotors are $31\frac{1}{2}$ inches, the H.P. main turbine rotor drum is $44\frac{1}{2}$ inches, and the L.P. rotor drums are 66 inches in diameter respectively. The blades range in length from $\frac{3}{4}$-inch in the H.P. cruising up to $7\frac{1}{2}$ inches in the L.P. main turbines. At full powers the shafts revolve at about 750 revolutions per minute.

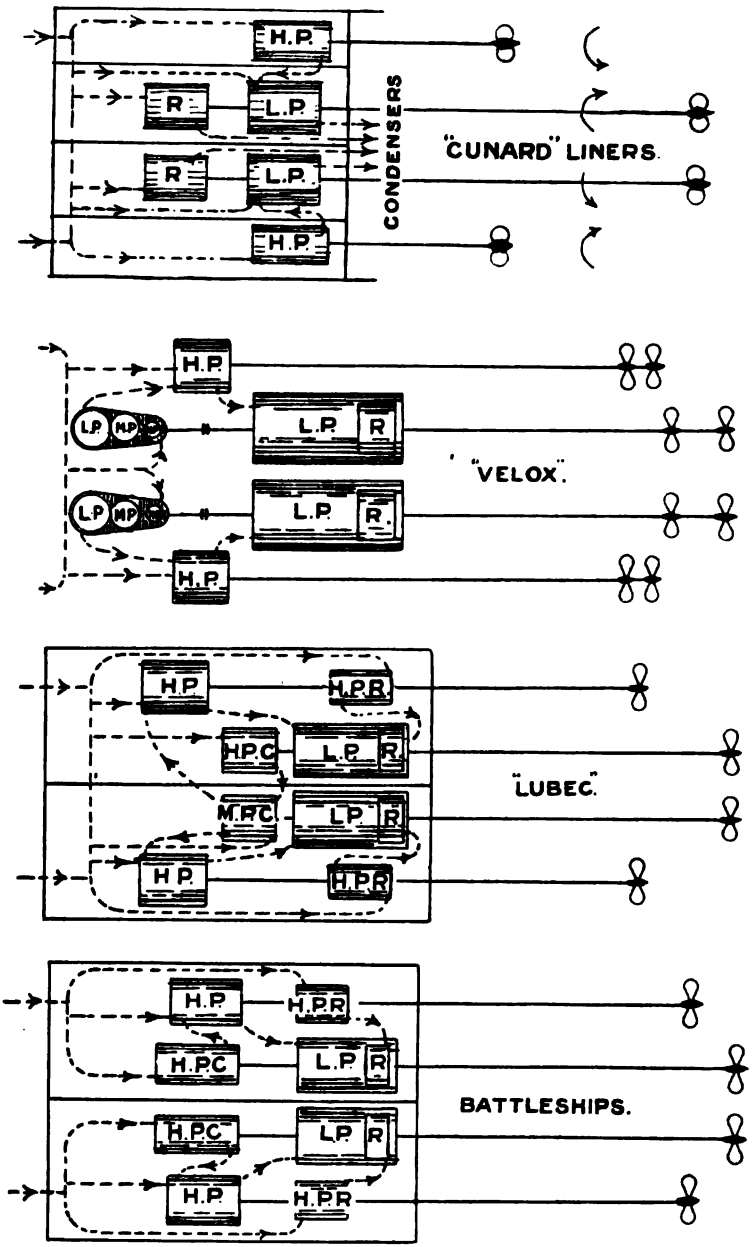


FIG. 368.—Four-shaft Arrangements of Marine Turbines.

Four-shaft Arrangement of Turbines.—Fig. 368 shows various arrangements of turbines for driving four propeller shafts, as fitted in vessels typical of certain classes. As in the case of the three-shaft arrangements an *idle* turbine (*i.e.* one not actually steam-driven) is free to revolve with its screw propeller, and both steam and exhaust ends of an idle turbine are connected temporarily with the exhaust, so that the rotor blades move in a medium offering a very low frictional resistance. The dotted lines show the general run of steam piping connections.

Cunard ss. *Mauretania.*—Four shafts; four screw propellers. One H.P. turbine on each wing shaft, and one L.P. turbine on each middle shaft. Each L.P. ahead turbine can be worked independently of the H.P. turbines. The middle shafts can be driven astern by astern turbines fitted independently on the L.P. turbine shafts.

H.M.S. *Velox.*—Originally fitted with combined reciprocating (three-stage expansion) engines, as shown, but these have now been replaced by cruising turbines. It was necessary to disconnect the reciprocating engines at high powers. Two screw propellers are fitted on each shaft, one in front of the other, to obtain the necessary blade area to prevent cavitation occurring. Only the middle shafts are reversible by means of manœuvring valves in connection with the L.P. and astern turbines.

German Cruiser *Lubeck.*—Two screw propellers originally fitted on each shaft, but since replaced by one on each. Turbines fitted in two engine rooms divided by mid-line bulkhead, through which cross connecting pipes pass for cruising turbines. Reversing turbines fitted on each shaft, one H.P. and one L.P., the latter being fitted inside the L.P. ahead casing.

British Battleships.—An improvement on the *Lubeck* arrangement, in that each engine room is independent of the other. Single solid screw propeller (blades not removable) fitted on each of the four shafts.

Combined Reciprocating and Turbine Engines.—In a French destroyer a *combined* arrangement of a *reciprocating engine*, for utilising the steam during its fall from a high to a moderate pressure, *and* of *turbines* for utilising the comparatively very low pressure steam exhausted from the reciprocating engine, is now under construction. The destroyer is fitted with Rateau turbines, and the combined arrangement is from Mons. A. Rateau's design. The economy of the combined system was tested successfully some years ago in the *Velox* and in a torpedo boat built by Messrs. Yarrow, but the arrangement described below has many advantages over the previous ones. The destroyer has a displacement of 430 tons, and a designed speed of 28 knots, and is propelled by a screw on each of three shafts. The middle screw and shaft is driven by a reversible three-stage expansion reciprocating engine, as shown in Fig. 367; but it should be pointed out that this figure is not drawn to scale, nor is it necessarily exact in its detail, for which material is not available at the time of writing. Each outer shaft is driven by a Rateau turbine, which is described later, of the impulse type.

The general system of the design is that at speeds less than 14 knots (250 revolutions, and at powers less than 900 H.P.) the piston engines alone supply the propelling power, but the exhaust steam from the L.P. cylinder traverses the turbines on its way to the condensers, so that they revolve at just such a speed that, although they do not assist the propelling power, the screws rotate in the water without sensible resistance, and thus the resistance of the vessel is diminished. Above 14 knots, the power given to the turbines provides for a necessary share in the propulsion. At 20 knots and over, boiler steam is admitted directly to the turbines, which then provide a large proportion of the propelling power. Running at the full speed of 800 R.P.M. the two turbines are designed to produce a total of 7050 effective H.P.; running at the full speed of 380 R.P.M. the piston engines are designed to develop 2150 I.H.P., thus making a total H.P. of about 9200 for the piston engines and turbines combined. This power should produce about $29\frac{1}{2}$ knots.

For "astern" working at low powers, the piston engines are sufficiently powerful, and steam can be exhausted directly from them into the condensers. Otherwise "astern" turbines are provided, which are distinct from the "ahead" turbines but enclosed, as in the Parsons system, inside the exhaust casings. The principal

difference between this system and the one used in the Yarrow boat is that the piston engine is designed to work at full speed, and no engine or turbine is then idle except the "astern" discs.

Another combined arrangement is fitted in the White Star ss. *Laurentic*. There are three shafts and screw propellers; the wing shafts are driven by reciprocating engines, of the three-stage expansion, four-crank balanced (naval) type, and the middle shaft by a turbine which utilises the exhaust. The efficiency of this arrangement is undoubtedly sound; but *for war-ships* a better arrangement combines on the same shaft turbines of the "impulse" (Curtis) and "reaction" (Parsons) types, in which "impulse" is used for the high-pressure stage or stages, and "reaction," absorbing the propeller thrust, for the final (low) pressure stage.

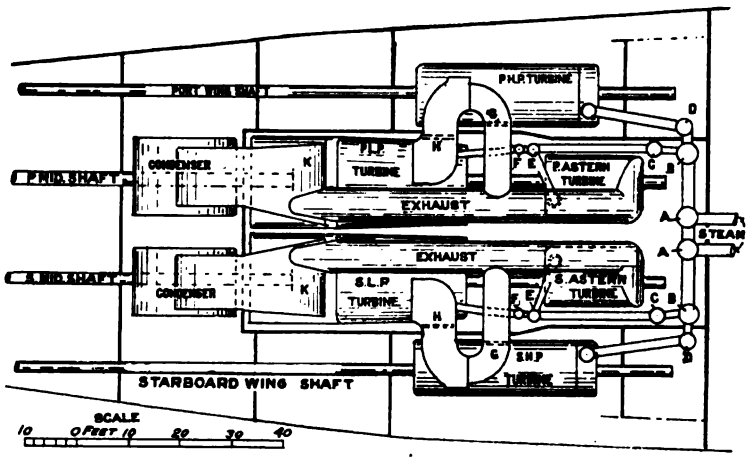
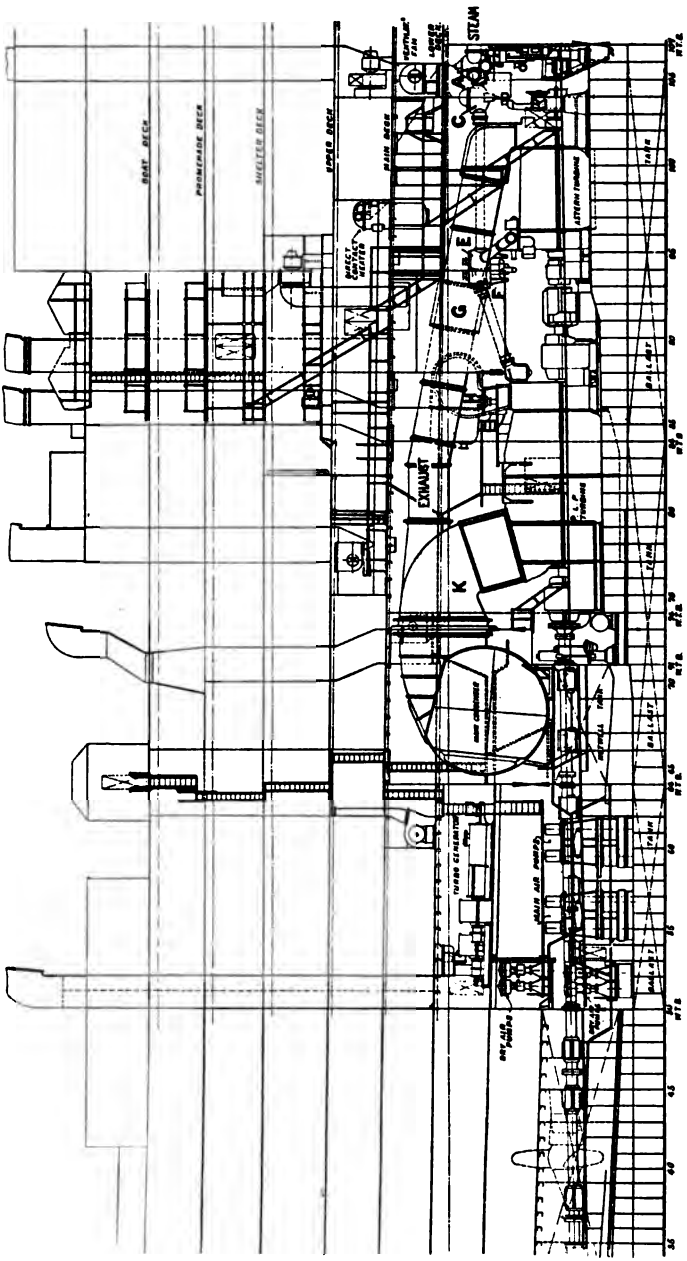


FIG. 369.—*Mauretania* Turbine Engine Rooms.

SS. "Mauretania."—The turbine machinery of this vessel and of her sister ship *Lusitania* would require a volume in themselves to adequately describe them, and it is only possible here to point out the principal features. Two figures, 369 showing a general outline, and 369A showing the general detail more fully, are given of the *Mauretania* engine rooms, and from these it is possible to follow out the scheme of arrangement. The letters denote similar parts in each of the views.

Steam enters from the boiler rooms through two valves, *A*, connected by a cross connecting pipe. The H.P. turbines are fitted



SECTIONAL ELEVATION.

FIG. 369A.

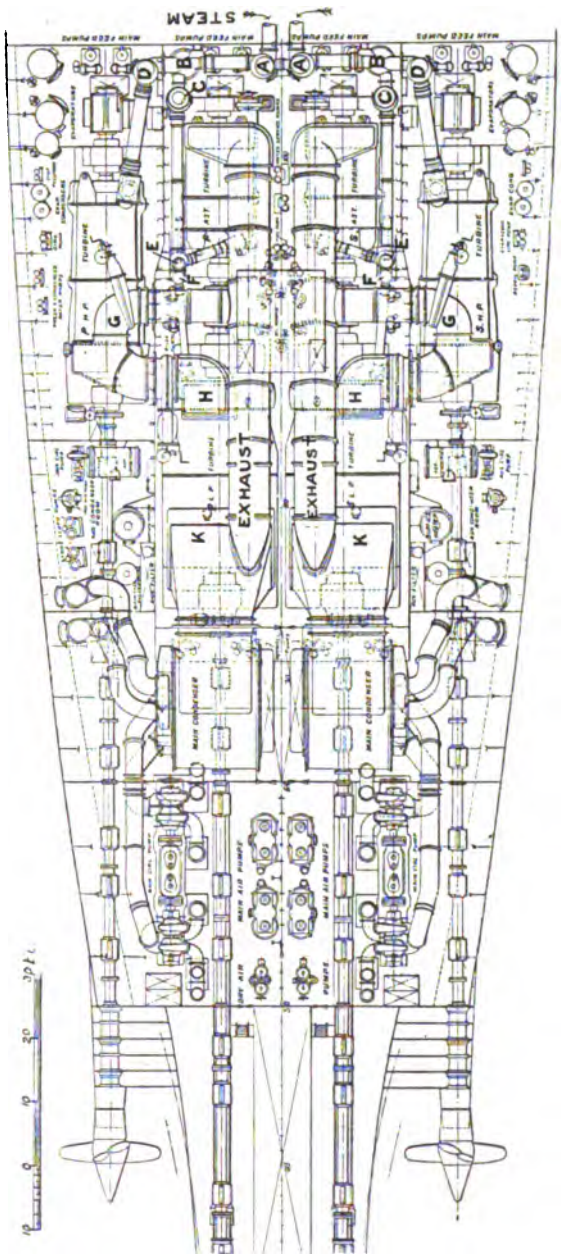


FIG. 369A.—General Arrangement *Mauretania* Turbines and Shafting.

in separate engine rooms on each side, and steam is admitted to them through separators, *B*, and shut-off valves, *D*. By means of sluice valves, *G* and *H*, near the exhaust end of the H.P. turbines, it is possible to work the H.P. turbines independently of the L.P. turbines by exhausting direct through the eduction pipes *K* into the condensers. The two midship engine rooms each contain a separate astern turbine and a L.P. turbine. Only the midship shafts can be driven astern, and when manœuvring (*i.e.* entering or leaving harbour, etc.) the steam is shut off from the H.P. turbines by means of the valves *D*, *G*, and *H*. Steam for manœuvring is then admitted through the stop valves *C*, which are usually closed when at sea to prevent leakage, through the manœuvring valves *E* for astern working, and *F* for ahead working (L.P. only). The thrust blocks are fitted ahead of the H.P. turbines on the wing shafts, and between the astern and L.P. ahead turbines on the middle shafts. The wing propellers are nearly 80 feet *before* the middle screws (not shown in Fig. 369A), and turn inwards. The middle screw propellers turn outwards. The main condensers occupy a compartment by themselves, and other separate compartments are occupied by the auxiliary condensers and main air pumps and circulating engines. The propelling machinery and its principal accessories thus spread over nine compartments, besides several auxiliary compartments.

H.M. Battleships.—The general arrangement is shown in Fig. 368 and, in fuller detail, in Fig. 370. Steam enters each engine room from the boilers by a separate pipe. When steaming at low powers, and the cruising turbines are in use, the valve *D* is opened and the power of the engine and supply of steam controlled by the valves *C*. For high powers, when the cruising turbines are not in use, the valves *C* and *D* are closed and the drains in connection with the inlet and exhaust ends of the H.P.C. turbines are opened to the condenser, and steam is admitted direct to the H.P. turbines through *A*, and, exhausting through the receiver pipes *H*, then drives the L.P. ahead turbine. For reversing, the valves *A*, *C*, *D* are closed and the drains at the inlet and exhaust ends of the H.P.C., H.P., and L.P. ahead turbines opened, and steam is admitted through the valve *B* to the H.P. astern turbine, which, exhausting through *E* into the L.P. astern turbine, reverses the midship shaft at the same time as the wing shaft.

The advantage of this system of arrangement over previous

arrangements, whether with two, three, or four shafts, is considerable. A glance at the diagram, Fig. 288, shows at once the possible increase of turning moment by the screws when the vessel is manœuvring. Working the steam in two stages of expansion, in separate turbines, gives considerable increase in economy when going astern, or entering and leaving harbour, and each engine room is entirely self-contained, being separated by a solid bulkhead. The general detail and method of operating the various valves will probably undergo modification, but it is anticipated that its essential

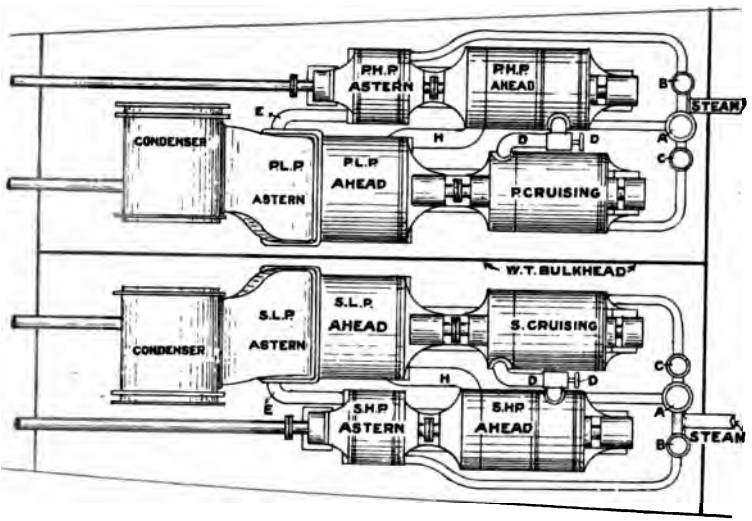


FIG. 370.—Battleship's Turbines.

arrangement for four shafts will continue in favour for Parsons turbines in much the same way as the general arrangement of twin-screw reciprocating engines has continued for many years past. It is also anticipated that wherever it is found possible to install four shafts instead of three, that four will be fitted, although a transverse bulkhead, or bulkheads, may be used in place of one at the mid-line.

Turbine Lifting Gear.—In Fig. 371 one of the L.P. turbines of the *Mauretania* is shown open for inspection. The upper half of the turbine casing has been lifted by means of a 30-B.H.P. electric motor. The guide columns *A* are 7 inches in diameter, and are bolted to the turbine casing and braced at the top or otherwise secured to the deck beams. The two columns at each end of the turbine are embraced by guide brackets on the

columns. Forged steel columns, *B*, can be inserted between the

FIG. 371.—Turbine Lifting Gear.

upper and lower casings when requisite to lift the rotor. For

lifting the rotor a cast-steel strap lined with white metal is slung round the shaft between the main bearing and the gland, and the slings are then connected with the lifting gear, and the rotor lifted and guided by the guide columns. Means are also supplied for lifting and transporting various parts, such as the caps of the main bearings, parts of the exhaust pipe, covers of thrust blocks, etc., and the lower parts of the main bearings can also be removed. The various arrangements vary according to ideas of those concerned, but the essential point in all these arrangements is the guide columns which prevent damage to the turbine and particularly its blading.

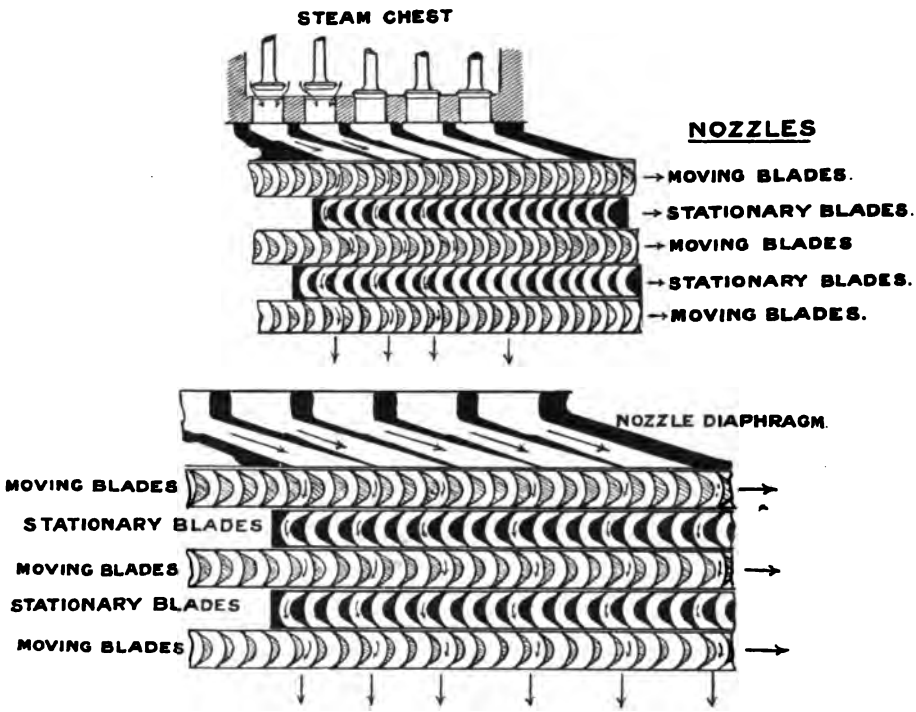


FIG. 372.—Diagram of Curtis Nozzles and Valves.

Curtis Steam Turbine. — The Curtis turbine is of the "impulse" type, but, as previously stated, it combines a certain proportion of "reaction" with "impulse," especially in the later stages. Steam is admitted to the first ring of moving blades, shown diagrammatically in Fig. 372, through valves and nozzles of the De Laval type. After acting on the first ring of moving blades,

the steam is redirected by a ring of stationary blades on to a second ring of moving blades, and for large turbines is again redirected, and so on for three or four rings of moving blades until the first stage of expansion is completed and the steam escapes into a space containing a diaphragm pierced with several nozzles arranged in groups. This nozzle diaphragm constitutes the beginning of a fresh stage of operation. Kinetic energy is again

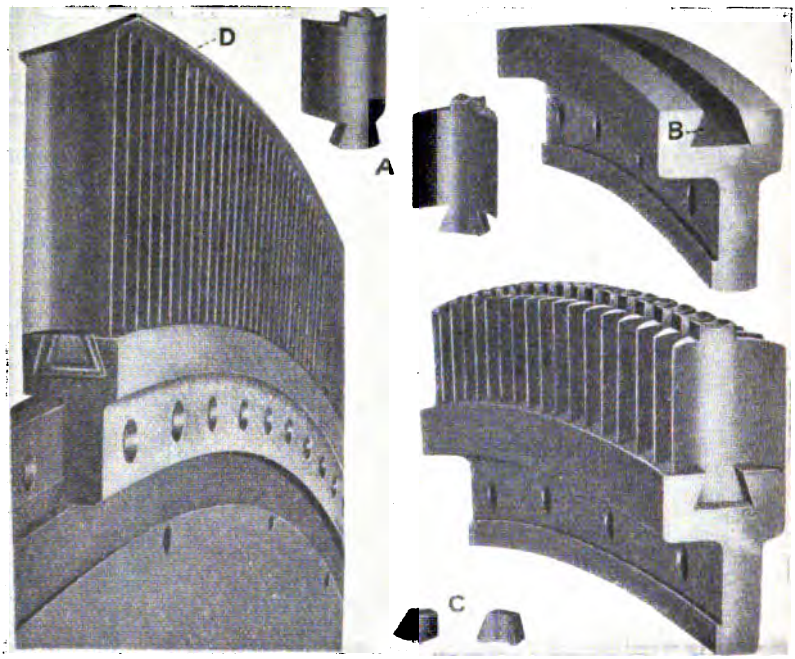


FIG. 372A.—Curtis Blading.

generated by a further fall of pressure in the nozzles, and energy of motion is imparted to successive rings of moving blades and the steam redirected by interposed rings of stationary blades. The steam escapes into a second chamber, ready for a third stage of expansion during its passage through nozzles fixed in a diaphragm as before. And so on for any number of stages as required. The Curtis system thus consists of several pressure stages with several rings of moving blades in each stage. The number of stages and moving rings is based on the velocity of blade and rate of revolution; and in the final stage the blading is sometimes fitted

on the Parsons system of continuous expansion. For short blades and small turbines, the blades are machined from the solid steel forming the rim of the rotor disc, but for large turbines with comparatively long blades the method of blading is shown in Fig. 372A. The blades are cut from drawn section wire to the correct shape *A*, and are then assembled in a dovetailed groove, *B*, with a filling piece, *C*, between each pair of blades. The outer ends of the blades are held by a shroud ring, *D*, and riveted over at the extreme end of the tip. Parts of rings, making up a complete ring,

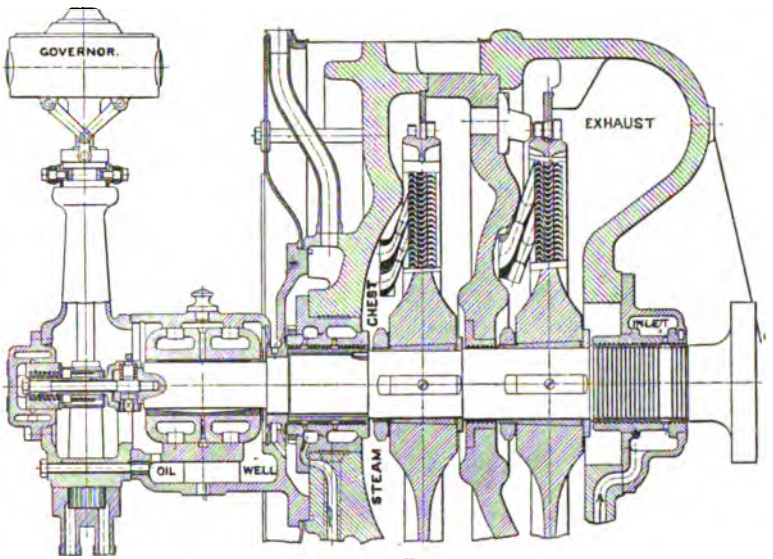


FIG. 373.—Two-stage A.E.G.-Curtis Turbine.

are riveted to the central disc, as shown in the figure. Stripping of blades is unknown with this method, and non-contact between the fixed and moving parts is ensured by the large tip clearance which can be allowed without reduction of efficiency.

Fig. 373 shows a part section through a two-stage turbine manufactured by the Allgemeine Elektricitäts Gesellschaft of Berlin (A.E.G. turbine) and rated at 850 kilowatts at 3000 R.P.M. when supplied with steam at 200 lb. pressure superheated to 600° F. The steam leaves the first set of nozzles at about atmospheric pressure, and expands from this to the condenser pressure in the second set. The steam-packed glands are made on the labyrinth system with "leak off" connections at the high-pressure end.

Curtis Marine Turbine.—Generally speaking, the marine system consists of two turbines working separate shafts, as in the ordinary twin-screw vessel with piston engines, and reversing is effected by fitting an astern turbine at the exhaust end of each ahead turbine. The arrangement is thus fairly compact and convenient, and is fitted in ss. *Kaiser*, 2000 tons, 6000 S.H.P. (A.E.G. Co.); U.S. cruiser *Salem*, 16,000 S.H.P., 24 knots, 350 R.P.M.; ss. *Creole*, 8000 tons, 8000 S.H.P., 17 knots, 235 R.P.M.; Japanese battleship and cruiser, 25,000 S.H.P. All the above, except the *Kaiser*, have been or are being constructed by the Fore River Company of Quincy, Mass., and the most recent for which details are available is the *Creole*, one of whose turbines is shown in Fig. 374. Each of the two turbines is 120 inches in diameter, with seven stages for "ahead" and two stages for "astern" working.

Referring to Fig. 374, the turbine consists of a cast-iron cylinder casing divided by diaphragms into seven ahead compartments and two astern compartments. In each compartment there is a separate wheel, with four rings of moving blades on the first and three on the remaining six of the ahead wheels. The wheels are mounted on a hollow shaft carried by self-aligning bearings. Where the shaft passes through the diaphragms, they are provided with bronze bushings having a small clearance, thus preventing appreciable leakage from one stage to another and lower one. Carbon stuffing in boxes prevents steam leaking *out* at the ahead end or air from leaking *in* at the astern end, where a vacuum exists during ahead working. The stuffing boxes are steam-packed, and are drained to the fourth-stage chamber. In the ahead steam chest there are twelve valves, each operating one of the nozzles for the first-stage wheel. For continuous running sufficient of the nozzles are opened to give the speed desired, and the ahead throttle valve is left open, thus giving full pressure in the steam chest. Nine nozzles will give full power, leaving three for overload. In the astern steam chest there are twelve nozzles, but only six are fitted with valves. For manœuvring, the nozzle valves in the steam chests are left open, and the speed is controlled by the throttle valves.

On each of the other six ahead stages two nozzle valves are fitted. Each valve closes one-fifth of the nozzle area of its stage. These valves are closed when running continuously at reduced speed, so that the proper steam pressure can be maintained in each

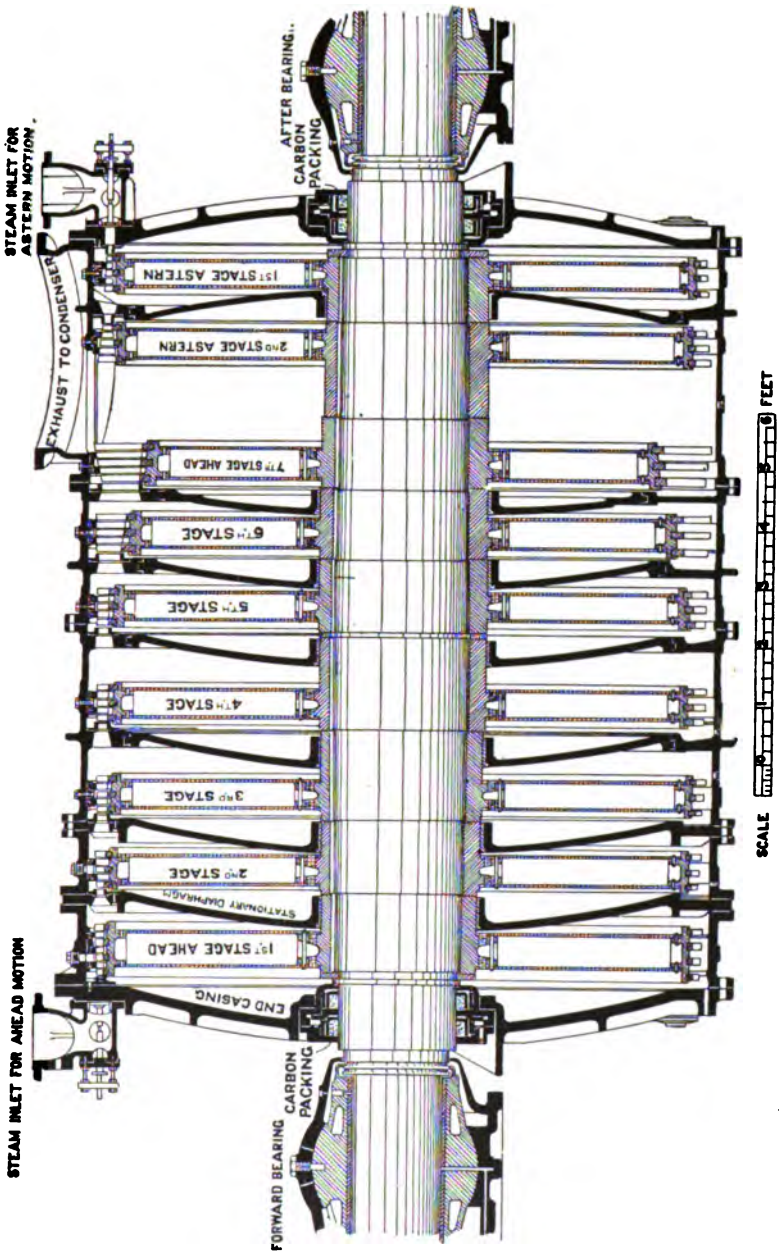


FIG. 374.—Curtis Marine Turbine (*Creole*).

stage. Drain pipes are fitted to connect each stage with the next lower one, so that the condensed steam in any stage can part with its heat in doing useful work. The exhaust chamber drains to the condenser, and the discharge is assisted by a small steam ejector. A common marine thrust bearing is provided for the forward end of the turbine shaft. As in the Parsons turbine, this bearing ensures the correct relative axial position of the rotor and fixed casing. Axial clearance of the blading is one-tenth inch in the first wheel, and increases up to one-quarter in the seventh. The length of the blades is increased towards the exhaust end, to allow for volumetric expansion of the steam.

In order to keep the pressure in the shell as low as possible the pressure distribution is arranged so that one-fourth of the available energy of the steam is expended in the first stage, and one-eighth in each of the other six stages. This requires the first stage nozzle to be of the expanding type, but all the other nozzles are of the parallel flow type.

On a four-hour run at full power an average speed of 16·7 knots was obtained; but the results were disappointing, because the propellers gave a low efficiency: and the stoking appeared to be indifferent, although the boiler efficiency (65·9 per cent) was not low compared with actual service conditions.

Reversing was satisfactorily accomplished, but various modifications and minor improvements are apparently necessary. Taking all things into consideration, especially that this was the first large vessel to be fitted with turbines in the United States, there appears to have been an assured success which may develop into a serious rivalry with the Parsons system. There is, however, much to learn, especially with regard to cruising and low speeds. If the cruising (Parsons) turbine can be abolished by adopting "impulse" turbines of the Curtis type, and the same economy of fuel, weight, and space can be obtained, as would appear to be the case, then the "impulse" turbine has a distinct claim for warship purposes; but, as pointed out elsewhere, in the author's opinion a combination of the two types appears to be the most economical practical solution of the cruising problem.

Curtis Vertical Turbine.—For electric generation and other purpose the Curtis turbine is usually of the vertical type, that is, the shaft is vertical. The general principle of construction and working is similar to the marine type. The British Thomson-

Houston Company make these turbines at their Rugby works; in the United States they are made by the General Electric Company of Schenectady, N.Y., and others: and in Germany by the A.E.G. Co. of Berlin.

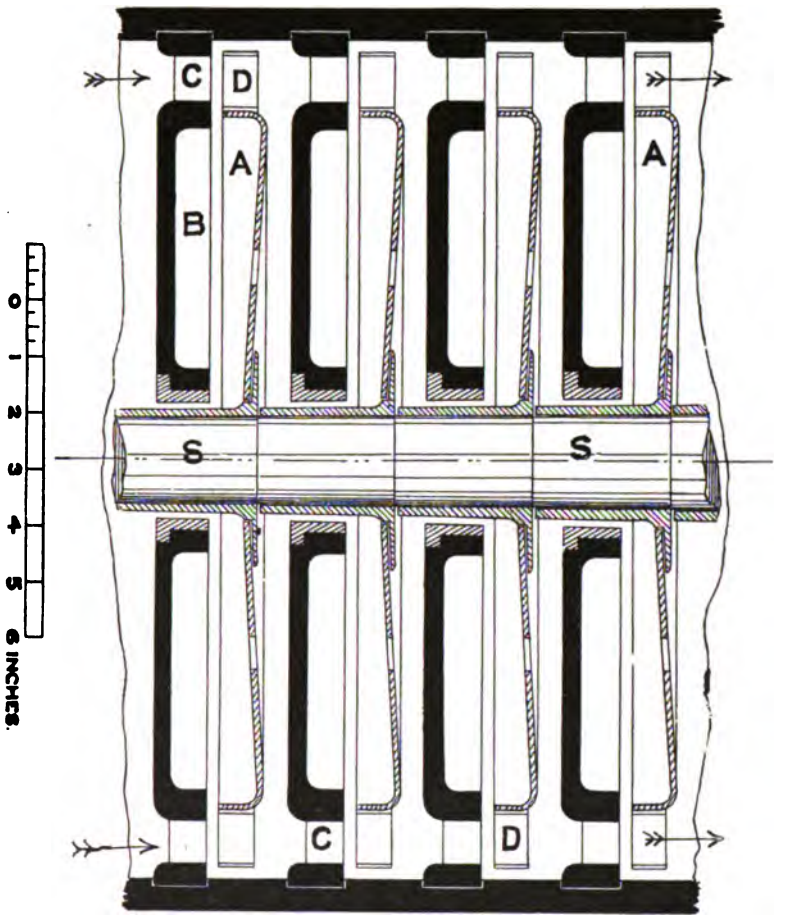


FIG. 375.—Rateau Nozzles and Rotor-Wheels.

Rateau Marine Turbine.—The Rateau turbine is of the "impulse" type, and there is practically no reaction. A short description of the arrangement of Rateau turbines, in conjunction with a reciprocating engine, has been already given.

Referring to Figs. 375 and 375B, it will be seen that there

are several rotating wheels, *A*, mounted on a shaft, *S*, and interposed between these wheels are diaphragms, *B*, fitted to the casing and

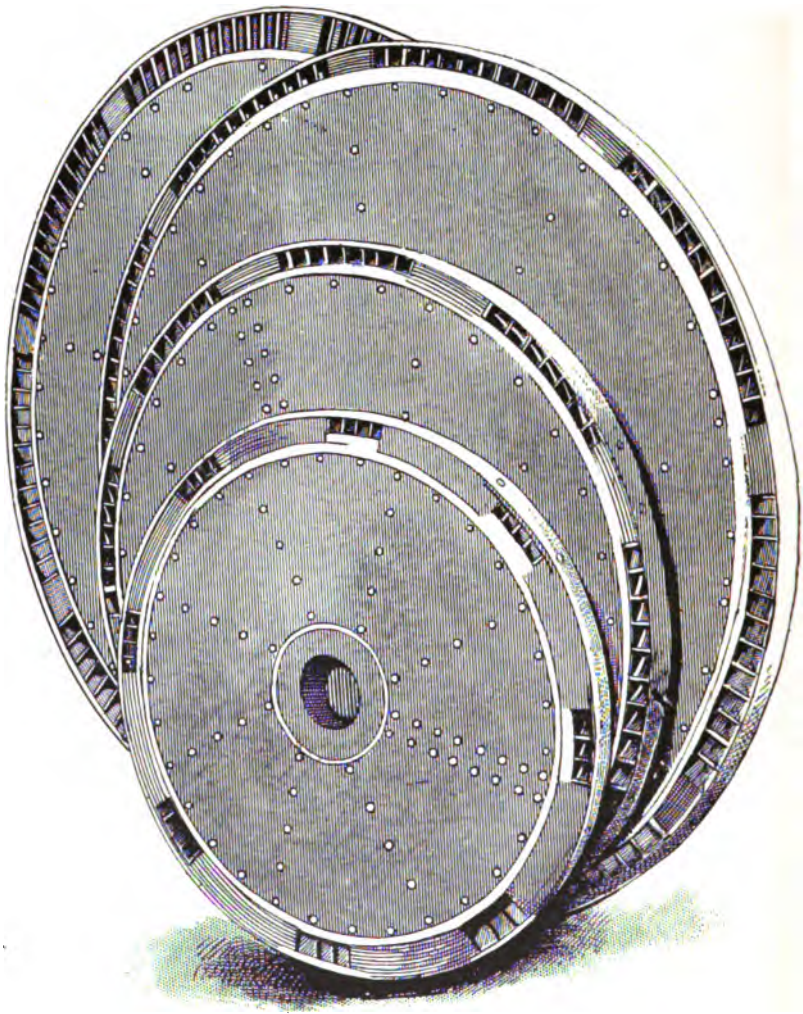


FIG. 375A.—Rateau Distributor Diaphragms.

containing nozzles, *C*, or steam distributors, of special shape. Successive drops in pressure take place in the succeeding rings of distributors. The moving wheels are made of thin sheet steel, and blades, *D*, of segmental circular form are riveted to the outer

circumference of the disc, and around the outer ends, or tips, of the blades a steel band is riveted to keep them in place and to maintain their correct spacing. The blades are fitted throughout the whole length of the circumference of the moving wheels.

Distributing vanes (or nozzles without valves) are fitted in part only of the circumference of the first diaphragms, thus somewhat resembling the De Laval turbine in regard to the partial admission of steam; but the passage through each successive diaphragm, as shown in Fig. 375A, is increased and set in advance of the preceding one, so that the steam leaving one moving wheel enters the succeeding distributor without encountering a solid wall, which would produce

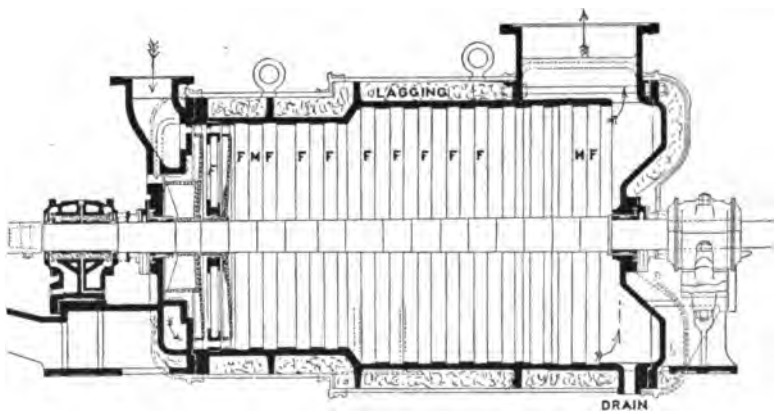


FIG. 375B.—Rateau Marine Turbine.

shock and consequent loss of rotative energy. For the final diaphragms, near the exhaust end of the turbine, it is necessary to employ the full number of openings possible within the circumference. Further, to allow for expansion of the steam, the length of the blades is increased gradually to give a larger passage-way, as shown in Fig. 375B, which shows the general arrangement of the high-pressure turbine fitted in a torpedo boat built by Messrs. Yarrow. For reversing, an arrangement of a series of two or three diaphragms and moving wheels is fitted near the exhaust end of the L.P. turbine, similarly to the Parsons and Curtis arrangements.

M. Rateau has given particular attention to the utilisation of exhaust steam, otherwise wasted into the atmosphere, and Messrs. Fraser & Chalmers, the English agents, have fitted up several turbines for this purpose with success. Engineers are indebted to

M. Rateau for his scientific and practical investigations into the flow of steam through convergent-divergent nozzles, and other matters intimately relating to turbine work.

Other Steam Turbines.—For marine propulsion the principal turbines are the Parsons, Curtis, Rateau, and Zölly. The Zölly turbine is under trial in Germany, and Messrs. Escher, Wyss & Co. of Zurich are working steadily at the problem of adapting their turbine to marine work; it bears some resemblance to the Rateau in its method of utilising the energy of the steam.

In the French Navy a modification of the Parsons turbine, by Brequet, as well as the Parsons turbine, is under trial, and many others have been proprosed for marine purposes.

For generating electricity and other general purposes the number of steam turbine systems is steadily growing, and, in addition to those already mentioned, the Hamilton-Holzwarth, Reidler-Stumpf, Kerr, Sturtevant, Elektra, and Union are perhaps the best known. Thus the turbine engine is making rapid progress in all the principal manufacturing countries, and for certain purposes for which it is more suitable is likely to supersede the reciprocating engine, unless its place is taken by some form of reciprocating internal combustion engine described in the next chapter.

Efforts have been made to combine the turbine principle with internal combustion, but, up to the time of writing, not with any great success, principally owing to the difficulty of dealing with the excessively high temperatures (3000° F. and upwards) at the admission end of the turbine. The difficulty is generally in relation to a suitable material for the blading and nozzles, and it is quite possible that a solution of the problem may present itself at any time, either by lowering the temperature by generating steam in the first instance, or by finding a sufficiently refractory material.

CHAPTER XXXV

INTERNAL COMBUSTION ENGINES

Introductory.—An elementary example of an internal combustion engine is a gun or cannon using some form of explosive as motive power. In this case the combustible is supplied, mechanically mixed with the necessary oxygen for its combustion. Detonation or ignition supplies the necessary local temperature to some small portion of the mixture and starts combustion, which may be either extremely rapid, or explosive, or comparatively slow, when it is termed slow burning.

The combustion generates an enormous relative volume of gas from the mixture. In the case of cordite the relative volume is about 700 at atmospheric pressure and at 32° F., and these 700 volumes are compressed almost instantaneously into only one volume behind the projectile. The pressure is still further increased by the rise in temperature, which is about 3750° F. for cordite, of the exploded or burnt gases; because such increment of temperature produces a corresponding rise in pressure when the volume in which the gases are confined is practically constant (or only slowly increased) in accordance with the law of gases (pressure varies as the absolute temperature when the volume remains unchanged). The total pressure thus sets up motion of the projectile and expels it from the gun, and the energy imparted to the projectile is in exact ratio to the work done and to the energy expended in the bore of the gun. The energy so expended is heat energy, and the gun is simply a form of heat engine which is governed by the same laws as any other heat engine, such as the steam and oil engines and the compressed air-engine in a torpedo.

In internal combustion engines which are used as primary motors for supplying power, a similar action takes place behind a

piston in a cylinder. The combustible, and the air which contains the necessary oxygen for completing the combustion, are supplied independently, and the relative quantities are regulated for each explosion; but as the combustion is comparatively slow, not instantaneous, it is not quite correct to term it an explosion.

Various combustibles are used in internal combustion engines, the principal being town gas (produced from bituminous coal) from the ordinary gas main, producer gas, industrial alcohol, benzol (produced from coal-tar), petrol, paraffin, and other comparatively light oils produced from petroleum, and heavy oils, such as petroleum, almost in a crude state. An oil engine does not differ in principle from a gas engine, but some method is generally adopted either for vaporising the oil into a gas before it is ignited and burnt in the engine cylinder, or for injecting the oil and atomising it under pressure. These methods are explained later, and the general principle on which internal combustion engines work will now be shown.

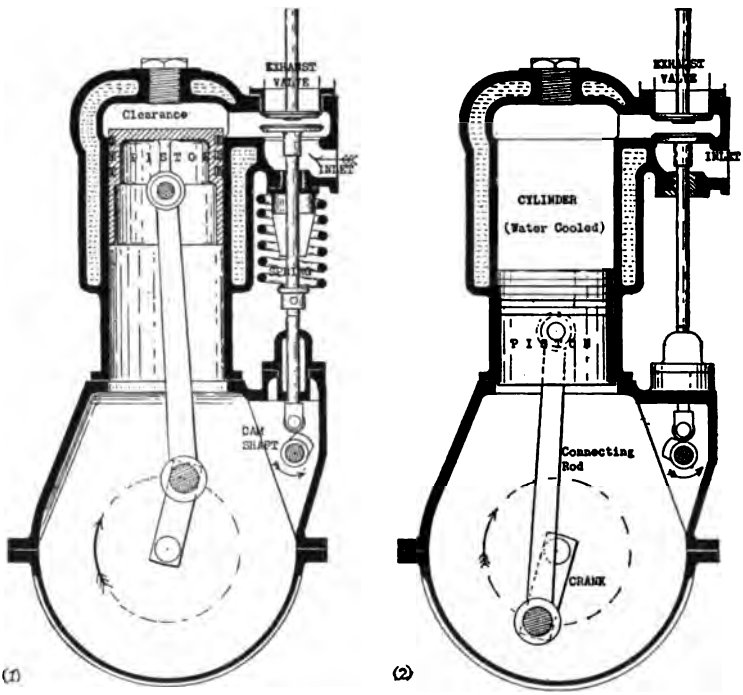
Beau de Rochas, or Four-stroke, Cycle.—This cycle, which refers to a series of operations consecutively performed and continuously repeated while the engine is working, was patented by Beau de Rochas in 1862; but no material progress was made until 1876, when the Otto-Crossley engine made its appearance. The cycle consists essentially of a single-acting engine, only one end of the cylinder being used to produce motion, with one working stroke in every four, corresponding to one reciprocation in every two revolutions of the crank. (In the ordinary double-acting steam engine there is a working stroke for every reciprocation of the piston.) For internal combustion engines the piston is usually of the trunk type, connected directly with the crank by a connecting rod, and therefore without a piston rod. The four-stroke cycle for a vertical inverted engine (Figs. 376 and 376A) is as follows, in order of the strokes of the piston:—

1. *Suction stroke,* during which the air is drawn into the cylinder by the downward movement of the piston through an *inlet* valve. In some engines the combustible gas is drawn into the cylinder at the same time, but in many recent oil engines the charge is usually introduced at some definite instant just previous to the completion of the next, the compression, stroke.

2. *Compression stroke,* during which the air drawn in during the previous stroke is compressed into the clearance space behind

the piston. For gas engines the cylinder clearance space is comparatively large, and the pressure only moderate, from 5 to 40 lb. per square inch; but for oil engines the clearance space is very small, and in the Diesel engine the pressure rises to about 600 lb. per square inch.

3. *Working stroke,* during which pressure is exerted by the



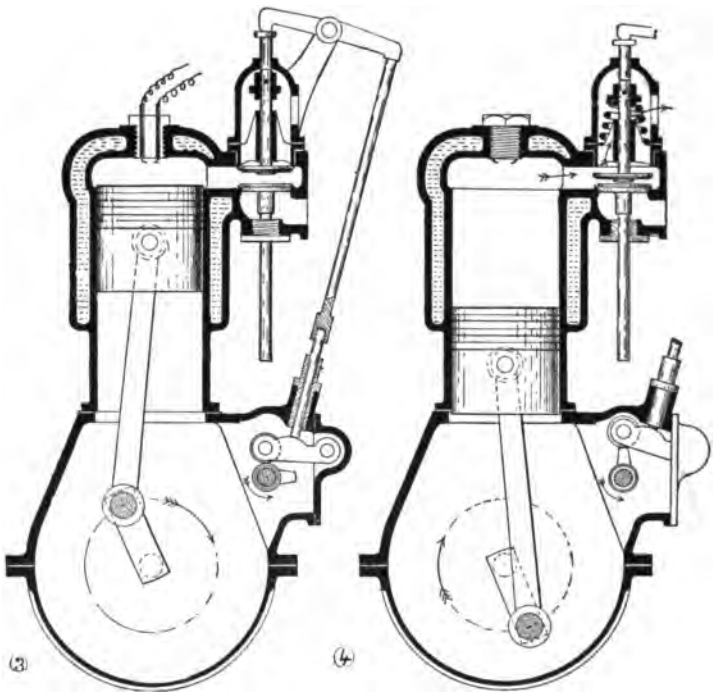
Inlet opens, *suction* begins. Inlet closes, *compression* begins.

FIG. 376.—Four-stroke Cycle.

gases on the piston, and work is done in giving motion to the engine. The charge is generally ignited just previously to the end of the compression stroke, but not so early as to retard or reverse the direction of motion of the shaft. The combustion produces great heat and temperature, which in turn produce great pressure on the piston which is pushed forward or downward; during the working stroke the gases expand, and continue to do work as the piston moves downward. At about 0·8 to 0·9 of the working stroke the

exhaust valve opens, and the pressure from the cylinder begins to make its escape, generally to the atmosphere.

4. *Scavenging stroke,* during which the products of combustion are expelled from the cylinder by the movement of the piston, but it should be noticed that complete expulsion is never attained, because some products remain to fill the clearance space in rear of,



Ignition, ready for *working* stroke. Exhaust opens, *scavenging* begins.

FIG. 376A.—Four-stroke Cycle.

or above, the piston. The exhaust valve is closed at about the end of the scavenging stroke, and the inlet air valve opened immediately after.

Since the expiration of the Otto patent in 1890 the four-stroke cycle has been very generally adopted for shore installations, because the mechanical difficulties of a double-acting engine are not yet entirely surmounted, although considerable progress has been made, especially with two-stroke cycle engines.

An indicator diagram taken from a cylinder working on the four-stroke cycle is shown in Fig. 377. This diagram records the pressure on the back of the piston only; the crank, or front, end of the piston is open to the atmosphere, and the pressure on it is atmospheric, or nearly so, and is represented in the diagram by the atmosphere line (14·7 lb. absolute pressure). The suction line naturally falls slightly below the atmospheric line, and the exhaust line naturally rises slightly above the same line. During compression the pressure rises from about 14·7 to 87·5 lb. without

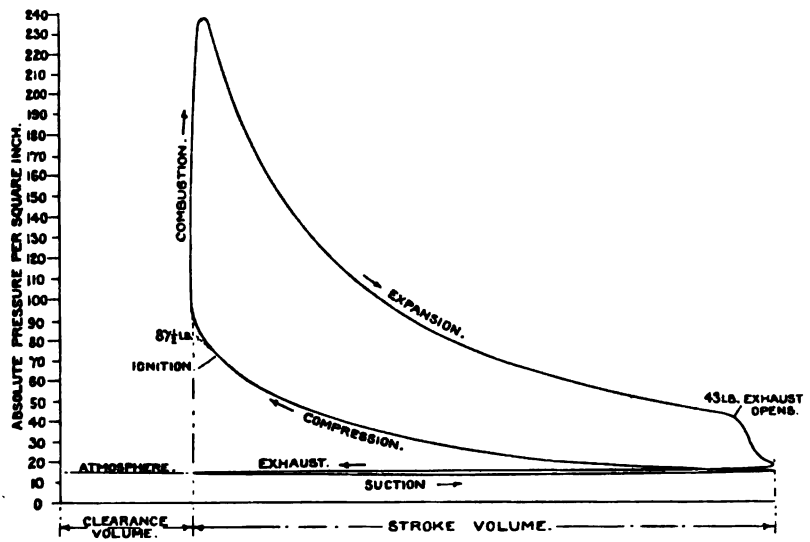


FIG. 377.—Indicator Diagram of Four-stroke Cycle.

ignition, but generally ignition occurs just before the end of the stroke, so that the pressure rises a little, as shown by the diagram near that point. During the working stroke the pressure increased by combustion rises rapidly to about 240 lb. and then falls during expansion, until the exhaust opens and allows the gases to escape and the pressure to fall to almost atmospheric. It should be noticed that the pressure attained at the end of the compression stroke (without ignition) is greater when the ratio of clearance-volume to the stroke-volume is less.

Cam or Half-speed Shaft.—In the better types of four-stroke cycle internal combustion engines the inlet and exhaust valves are

mitre seated, and generally spring controlled and operated by the action of the cams on the cam shaft; but some makers still allow the inlet valve to be automatically opened by the suction created by the piston, but as more is now known of the theoretical and practical working of the engine, this method is becoming unusual, and rapidly giving way to the more accurate method of mechanical operation. The position and arrangement of the inlet and exhaust valves should be relatively convenient, the one to the other, for the functions they fulfil.

The cams fitted on the half-speed shaft bear on rollers fitted at the lower ends, in a vertical engine, of the prolonged valve rods. In the Diesel and some other engines the valves are operated by a rocking lever whose opposite end is attached to the valve spindle. The cams, rollers, pins, etc., which form the working surfaces are nearly always case-hardened.

A cam shaft is generally unnecessary to a two-stroke cycle engine, but in the four-stroke cycle it is revolved at one-half the rate of the crank shaft; the relative rate of revolution of the two shafts, cam and crank, being governed by toothed wheels or by some form of worm-screw and wheel gearing.

Six-stroke Cycle.—Recently some experiments have been made with an engine working on a six-stroke cycle, in which both the exhaust and suction strokes are duplicated, and thus a purer charge of air is obtained for the compression stroke together with a slightly greater weight of air admitted to the cylinder, because of its lower temperature and consequent greater density. Where weight and space are of no particular importance the slightly greater thermal efficiency obtained by a six-stroke cycle may have some commercial value, but for marine purposes one working stroke in every six of the piston is too small a proportion to compete successfully with the four- or two-stroke cycle, in which, in addition, the engine may be made double acting. Although the six-stroke cycle may show a gain in thermal efficiency, it is evident that the mechanical efficiency must be less than that of an engine working on a fewer cycle and consequently with fewer working parts.

Two-stroke Cycle.—In this arrangement, first proposed and used by Mr. Dugald Clerk, every forward, or downward, stroke is a working stroke; and during every return, or upward, stroke the charge of air or mixture is compressed behind, or above, the piston ready for the working stroke. There is, therefore, one working

stroke for *every* revolution of the crank; in place of one working stroke for every *alternate* revolution, as in the four-stroke cycle.

The action in the two-stroke cycle is as follows:—On completing from 0·8 to 0·9 of the working stroke, the piston uncovers an exhaust port, Fig. 378, and thus allows the products of combustion to escape. A little later in the same stroke the piston uncovers another port which admits air, slightly compressed, generally to about 5 to 10 lb. above atmospheric pressure; suitable shaping of a baffle on the end of the piston directs the incoming air, so as to expel, as far as possible, the products of combustion, and to take their place. Shortly after the beginning of the return stroke, the air inlet and exhaust outlet are successively closed by the movement of the piston; compression now commences, and the cycle of operations begins for a fresh working stroke.

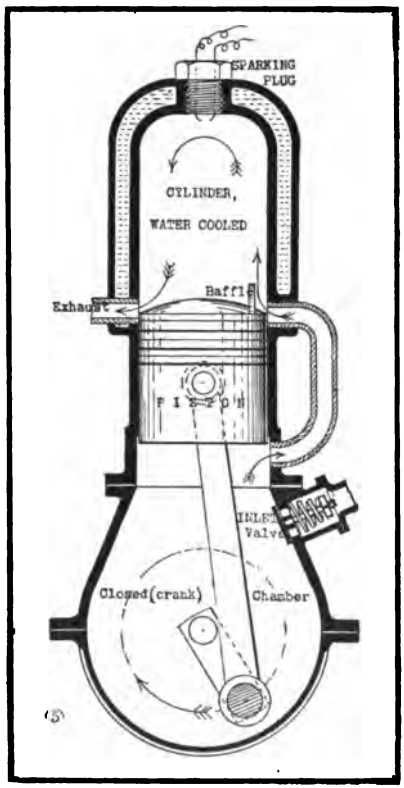


FIG. 378.—Two-stroke Cycle.

Various arrangements are adopted for producing the pressure necessary for scavenging. The most usual is where the crank chamber is enclosed; on the return, or up, stroke, air is drawn into the crank chamber through a non-return valve, and the down-stroke of the piston compresses this air, and expels some of it into the cylinder through the inlet port, or channel, already mentioned. A higher pressure, under proper direction, renders the scavenging more effective.

In Fig. 379 indicator diagrams are shown, taken from opposite sides of the piston of a two-stroke cycle engine. The pump diagram, obtained from the front or lower side of the piston, is shown on a larger scale for convenience of illustration. On the back of the

piston there is neither a suction nor an exhaust stroke, and consequently the pressure at the beginning of the compression stroke is slightly above atmosphere. The pressure accumulated (without ignition) is therefore slightly higher than in a four-stroke cycle, (assuming the proportion of clearance to stroke volume to be exactly the same). The pressure caused by combustion does not usually rise quite so high, because the charge is weaker than in the four-

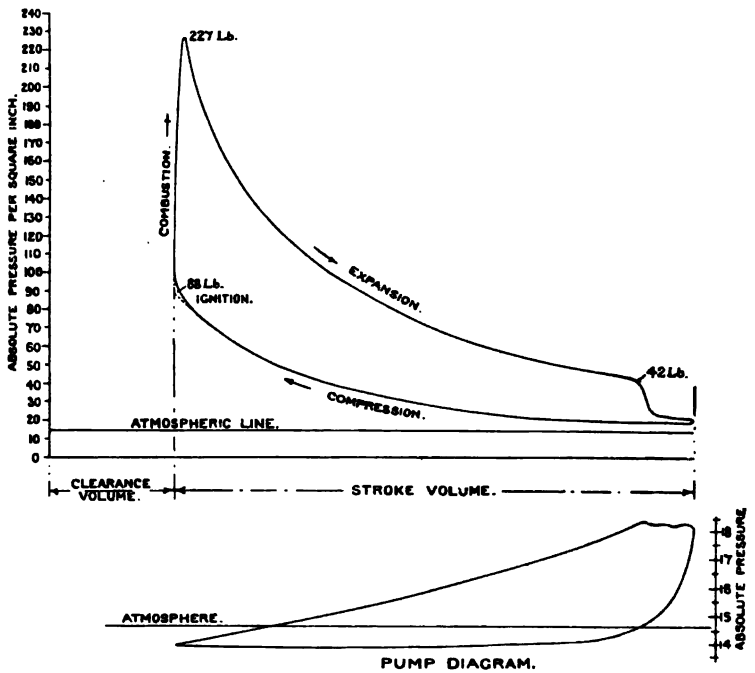


FIG. 379.—Indicator Diagram of Two-stroke Cycle.

stroke cycle; this weakness is due to less efficient scavenging and the presence of a greater proportion of nitrogen.

During expansion the pressure falls according to much the same curve as in the four-stroke, but not to quite the same extent, because the cylinder is slightly warmer and tends to maintain pressure during expansion.

On the pump side of the piston the suction stroke shows less than atmospheric pressure for most of its length, but at the beginning of the stroke the pressure, above atmospheric, produced by compression during the previous stroke and not utilised in scavenging, falls

quickly to atmosphere and below. The pressure accumulates steadily until the scavenging port is opened, when it remains almost constant until the end of the stroke, and then falls rapidly at the beginning of the suction stroke.

In *calculating the mechanical efficiency* of internal combustion engines the area of the pump diagrams of two, four, six (or any number) stroke-cycles represents a loss. It is usual, however, to deduct this area from the working area when calculating the I.H.P., and to consider the I.H.P. thus obtained as the true working I.H.P.

It might appear from the above description that twice as much power is obtainable from a two-stroke cycle engine as from a four-stroke engine, having the same sized cylinder, and under other similar conditions of working. In practice, however, such is not the case, and only about 60 to 80 per cent increase of power is obtained. Some part of the working stroke is inoperative (*i.e.* the part after exhaust begins), and also some negative work is done in compressing the air required for scavenging. The advantage for naval purposes is, however, very great, and even if it is determined to adopt the same *effective* stroke, thus increasing relatively the actual stroke of the engine, the saving in weight and space is very considerable. The thermal efficiency is perhaps lower owing to the difficulty of effective scavenging, but considerable improvement has recently been made in this direction.

A double-acting engine, of which there are now some building, necessarily works on the two-stroke cycle, and the same remarks apply as above. Of course, a satisfactory double-acting engine enables twice the power to be obtained on only a very moderate increase in weight. It involves a piston rod, crosshead and guide, and a consequent increase in height (or of overall length for a horizontal engine); and difficulties arise in keeping the stuffing gland tight with the high temperatures used, and in providing some separate means for compressing the air necessary for scavenging. These difficulties are principally mechanical, and therefore may reasonably be expected to be overcome as experience is gained.

Gas Engine.—The proximity of the town-gas main led to frequent adoption of ordinary lighting gas for power purposes, but only about 10 per cent of the total heat energy in the coal used for producing this gas is supplied through the main, while the remaining 90 per cent is left at the gas works principally in the form of coke.

For gas engines of this kind about one part of gas, consisting principally of hydrocarbons of the form CH_4 and C_2H_4, is mixed with from eight to twelve parts of air and the mixture admitted to the cylinder at each working stroke, where it is ignited or fired by some one of the several forms of ignition which are described later. The pressure thus generated forms the motive force, and the energy of motion is thus directly derived from the heat generated by the combustion.

Producer Gas Engine.—The high cost of town or illuminating gas as fuel for gas engines led to the adoption of a cheaper form, known as producer gas, which, although unsuitable for lighting purposes, is more suitable for the production of cheap power. There are two forms of producer—one of which works under a plenum, or excess above atmospheric pressure, and the other under suction or partial vacuum. The air, which is limited in quantity, is passed through the burning fuel, and CO, carbon-monoxide, is produced by partial combustion as well as other combustible gases, such as hydrogen and inorganic hydrocarbons. These gases form the basis of the combustible used in the engine cylinder. Pressure and suction producers differ only in detail and not in principle, but for marine purposes the suction type appears to be more suitable, and it is generally used on land in preference to the pressure type. The Dowson, Brunner-Mond, Campbell, Crossley, Capitaine, and Boutillier producers are some notable examples of the many in existence, and the description of one of them may probably be sufficient for the proper understanding of all. Improvements are being made daily, and it is quite possible that engines driven by producer gas may compete successfully with oil engines for marine purposes in the near future, principally because of the limited supply and increasing cost of oil.

Suction Producers.—The *Campbell Producer* consists principally of a steel chamber, *B*, lined at its lower part with brickwork, and fitted with a fire grate. The fuel, preferably anthracite peas of $\frac{3}{8}$ to $\frac{5}{8}$ inch in size, is fed into the top of the chamber through a hopper, *C*, which is kept nearly full when the apparatus is working, so that the fuel prevents the escape of gas when the hopper is opened.

Steam and air are fed into the space *D* below the grate, and, combining with the fuel, several actions and reactions take place in the lower layer of the fuel until the gases, evolved by the interaction of the fuel, air, and steam, pass away (as shown by the arrow heads)

into a pipe *E* leading into the *washer F*, and are there discharged below the water-level.

The gases are cooled in the washer, and tarry products, carried over with them, are there removed; these products floating on the surface of the water are run off automatically, or as required, through the drain *G*, which from its relative position maintains a constant water-level in the washer.

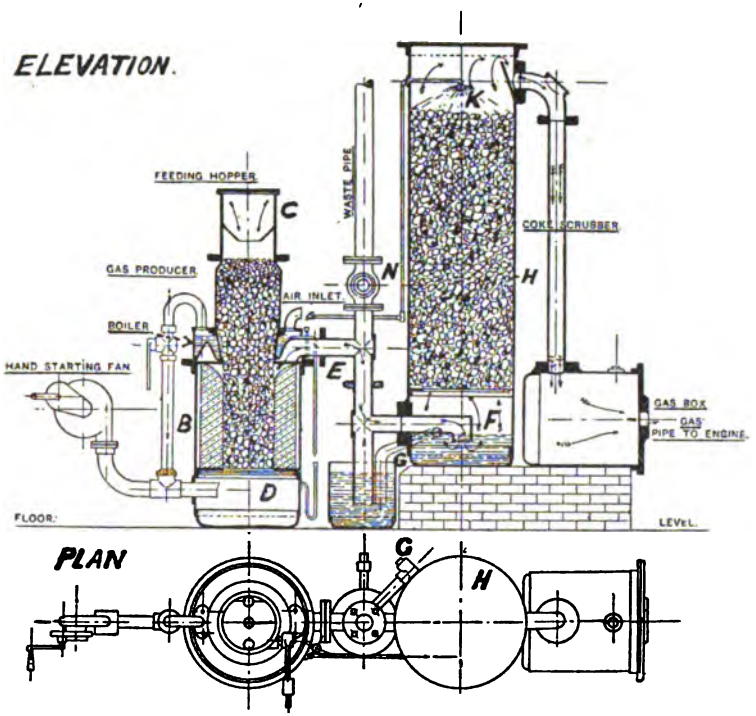


FIG. 380.—Campbell Suction Gas-Producer.

The gases now pass upwards inside a chamber, *H*, through the *scrubber*, which consists of a series of wooden trays in some types, but in the Campbell apparatus is generally of coke; the upward stream of gases is met by downward streams of water sprayed into the top of the scrubber at *K*. The effect of the scrubber, and the water spray, is to remove ammonia gas (which is generally present in moderate quantity, and formed by the interaction of the nitrogen contained in the air and hydrogen dissociated from the fuel and steam) and other impurities, such as sulphur and dust.

An analysis of producer gas gives results somewhat as follows, by volume and weight:—

	Dowson Producer.		Capitaine Marine Type.
	Volume.	Weight.	Volumes.
Hydrogen, H	18·73	1·67	14
Marsh gas, CH_4	0·31	0·22	1
Olefiant gas, C_2H_4	0·31	0·38	...
Carbon monoxide gas, CO	25·07	31·36	25
Carbon dioxide gas, CO_2	6·57	12·91	6
Oxygen, O	·03	·04	...
Nitrogen, N	48·98	61·27	54
Total	100·00	107·85	100

In Fig. 381 a sectional elevation is shown of a *Crossley Producer* adapted for using bituminous coal. Two feeding hoppers are

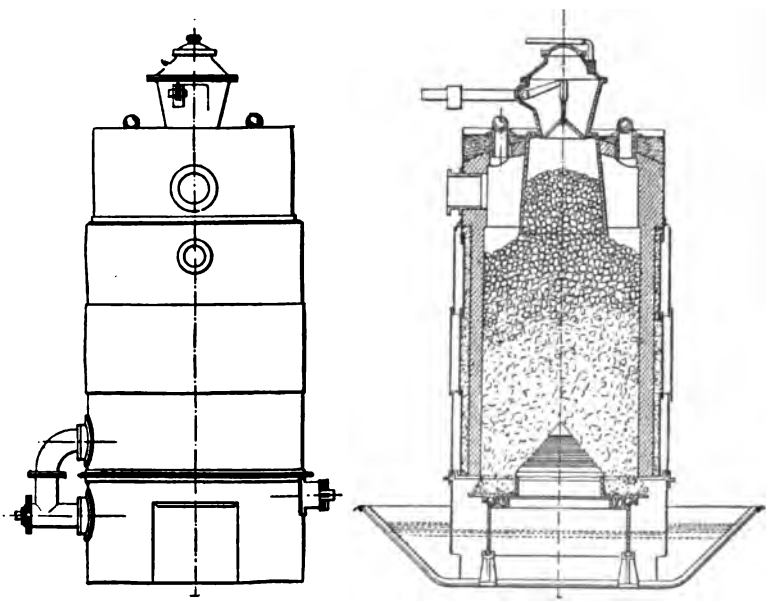


FIG. 381.—Crossley Suction Gas-Producer.

provided in the larger sizes, so that they can be filled alternately at intervals of about one hour. At the bottom of the producer there

is a water lute to enable the ash and clinker to be withdrawn without interfering with continuous working. The grates can be rotated, which saves poking and slicing. The gases on leaving the producer enter a *saturator*, where they are cooled by employing some of their heat to superheat the steam and air during its passage to the producer, and also to raise steam from the water surrounding the saturator. (The upper part of the Campbell producer performs a similar duty, and is called a *boiler*.) The gases on leaving the saturator have a temperature of about 212° F., and enter the washing apparatus where they are cooled to about atmospheric temperature. By means of a cooling tower and a circulating pump, the same cooling water is used over and over again. The gases next enter a centrifugal *tar extractor* (particularly necessary for bituminous coal), and pass on to a *sawdust filter*, where remaining traces of moisture and tar oils are removed before reaching the engine. The consumption of common slack coal per I.H.P.-hour is generally below 1 lb. in conjunction with a Crossley gas engine.

The more general and efficient method of dealing with bituminous coal in producers is to employ two or more producers in series; the gas generated in the first being passed through the second, which has already been entirely or partially reduced to coke. At necessary intervals one of the producers is charged with fresh fuel, and by changing over the connections and air inlets the producer last charged becomes the first of the series. In this way the tarry products distilled over from the first are entirely burnt in the second (and third, if fitted) producer. The amount of water and steam required is also much reduced by the series system, and for this reason it appears to be best adapted for marine purposes.

It should be noticed that the better classes of suction producers are constructed to work on the sound economical principle of utilising the waste heat otherwise carried away into the washer and scrubber, and the hot gases which eventually leave the producer at a temperature of about 200° F. are generally used to heat the air and steam supplied for combustion to the fuel.

When first starting the apparatus, a piece of lighted waste or other material is thrown into the ashpit, and a small blower, worked by hand, supplies the furnace with a draught. About twenty to thirty minutes are required to produce a good quality of gas in a producer of from 60 to 100 I.H.P., and until the requisite consistency of the gaseous products is obtained the cock *N*, Fig.

380, is left open, and the products allowed to escape into the open air. The proper quality can be recognised from the flame produced at a gas jet on the exit pipe. At very light loads the fire dies down, and the temperature is insufficient to produce good gas; it can, however, be maintained by partially opening the exhaust cock and allowing the gases to escape, but of course this is not an economical method because the escaping gas is wasted. Probably the most difficult matter with suction producers is the proper regulation of the air-supply for varying speeds and powers, but with a regular rate there is no apparent difficulty.

With a non-coking coal, the necessary area of fire-grate is about one-twentieth of a square foot per I.H.P.; but when coking and clinkering occur, about one-twelfth of a square foot is required, compared with about one-seventh for a natural draught steam boiler burning 15 lb. of coal per 1 square foot of grate. The thickness of the fire is generally maintained at about three times the $\sqrt{(\text{transverse section of the producer})}$.

The loss from cleaning fires when the producer is continuously in use is considerable, and is similar in character to that in the furnace of a marine steam boiler. During the period of cleaning and restarting the gas producer is thus out of action, but on this subject much has yet to be learnt of the practical value of the various methods used to overcome this difficulty; so far, continuous working for long periods has been accomplished by using anthracite and non-coking and non-clinkering coals, and some makers claim to have obtained good results with bituminous and slack coals.

The greatest objection, which is apparently almost insuperable, for marine work is the replenishing of the water used to produce the steam admitted to the fuel, and other wastages in the washer and cooling water used for the engine cylinders; it is quite possible, of course, to conserve much of the latter by using a surface cooler and by cleansing, but the steam waste is stated in one plant to be something like 10 lb. per horse-power hour or about one-half that of a non-condensing engine. The pressure required for this steam is low, not much greater than that of the atmosphere, and under this condition a means may be possibly found to overcome the usual difficulties either of maintaining a large supply of fresh water, or of preventing the formation of scale in ungetatable places when sea water is used in place of fresh.

The principal advantage of a suction over a pressure producer

is that the engine draws into its cylinder during the suction stroke the necessary quantity of gas for each working stroke, but the pressure sytem appears to be necessary for all but the last one of the series.

For small powers the weight and space required for a suction gas producer and engine is about the same as those for a steam plant of the same power, but there may be considerable saving with large-powered gas engines and producers. The fuel consumption is also less, and in the larger powers is only about one-half.

Efficiency of Suction Producer and Engine.—In a suction plant there is a limited supply of air and steam which combines with the fuel, and results in a combustible mixture of gases with less heat value than that of lighting gas per cubic foot of the mixture. But as more of the original fuel is converted into combustible gases, with less waste of material, a greater quantity of heat is obtained per 1 lb. of fuel expended in the producer. Pure carbon possesses a heat value of about 14,500 b.t.u. per 1 lb., and in the suction producer nearly 80 per cent by weight of this is converted into CO, carbon-monoxide, by the addition of a limited quantity of air; hydrogen is dissociated from water vapour, and adds considerably to the heat value; and the usual hydrocarbon gases are distilled off as in the production of lighting gas. These latter gases may vary the heat value in actual practice by their dissociation and association under varying conditions of temperature in the engine cylinder, but the net result is a large gain in combustible energy.

The proportion of the heat value of the fuel which is transferred to the working substance in any heat plant, whether steam or producer gas, constitutes its thermal efficiency. This efficiency for a suction producer is stated to be nearly 90 per cent, while for a steam boiler it may be as high as 80 per cent; allowing for differences of working under practical conditions, the relative efficiency of a suction producer to a steam boiler is about five to four.

The heat value of producer and other gas is generally stated in b.t.u. per 1 cubic foot of the gas at 60° F., and under the normal atmospheric pressure of 14·7 lb. per square inch. This value varies from about 150 for Dowson, Mond, and other gases to about 650 for ordinary lighting gas, but the value depends on the sort of fuel

used and the method of production. Only about $4\frac{1}{2}$ cubic feet of town-lighting gas are obtained from 1 lb. of coal compared with about 63 cubic feet of Mond gas, and thus more than three times as much total heat is obtained per 1 lb. of fuel from the latter as from lighting gas; actually as 9450 is to 2925 b.t.u. The cost of the fuel used in the production of Mond gas is also less, frequently only one-half. Very few gas producer engines require more than 1 lb. of fuel per I.H.P. hour, and many of them about 0·8 or less. Compared with lighting gas, that from a producer is comparatively slow burning; this is a useful property, as it allows a fairly high compression, greater range of expansion, and consequent economy. It also avoids violent shocks and produces smoother working, greater elasticity, and more constant torque.

Spirit and Petrol Engines.—These are gas engines in principle, because the spirit or petrol when admitted into the cylinder is either in a state of gas or in a rapid process of conversion from a liquid before ignition or firing. Petroleum spirit or petrol has a very low flash-point, and ignites easily at ordinary atmospheric temperatures and pressures. Roughly speaking, the flash-point and boiling-point of spirits and oils rise with the density or specific gravity. Some vapour is given off both by spirits and oils at all normal temperatures, but the quantity rapidly increases as the temperature approaches the flash-point; to this property is attributable the convenience of petrol for internal combustion engines, and also the danger of low flash-point oils or spirits for naval purposes.

The petroleum spirit used for ordinary motor cars has a flash-point below 73°, its density lies between about 0·67 and 0·72, and the boiling-point lies between 145° and 200° F. One gallon of fresh water, density = 1, weighs 10 lb., and therefore one gallon of petrol or spirit of density 0·70 weighs 7·0 lb. (For the flash-points, etc., of various fuel oils, see Chapter XII.)

Benzol.—Commercial benzol is produced from coal tar, and is a colourless liquid. When this liquid is heated to 212° F. about 90 per cent distils over, and it is therefore called *90 per cent benzol*. Distillation begins at about 177° F. Its density when purified is about 0·88, or, in other words, an imperial gallon of it weighs 8·8 lb. compared with 7 lb. for petroleum spirit. Its freezing-point is below that of fresh water, but by adding alcohol or gasolene its freezing-point can be lowered to about 0° Fahrenheit. It mixes

freely with gasolene, alcohol, and other hydrocarbon liquids, but it will not mix with water. Its heat value, by volume, is almost as great as that of petrol or petroleum, and, by weight, is about 18,000 b.t.u. per 1 lb. For internal combustion engines a compression up to 220 lb. per square inch can be used for benzol, and consequently a higher thermal efficiency than with gasolene or petrol is possible. A mixture with alcohol up to a proportion of 10 per cent cheapens the fuel without detracting from its desirable characteristics; and because alcohol has a future as a fuel for engines, it is probable that benzol may be selected as a denaturant for preventing fraud on the Revenue. By substituting benzol for petrol, gasolene, or benzine (petroleum spirit), almost all four-stroke cycle engines can be worked at half the existing cost of fuel, but for small engines a special carburettor is required.

Oil Engines.—The safest plan for naval purposes is to use *heavy oil*, with a flash-point above 150° F. The gas producer engine is, however, a probable competitor for naval purposes, and experiments are being made in this direction. The difference between using gas and oil is that the oil is gasified or vaporised either before or during admission of the fuel to the cylinder, and generally before ignition and combustion begin. In other respects the engines and their internal working are identical in principle.

Combustion.—In the complete combustion of *liquid* fuel, or hydrocarbon, there are three distinct processes: (*a*) atomisation, (*b*) vaporisation, and (*c*) carburetion. These three processes may or may not be amalgamated into one apparent operation, but in any case complete combustion without leaving a residue can only be produced by their co-ordination, whether the fuel is burnt under a boiler or in an internal combustion engine.

For atomising, some form of pulverising force is necessary, and may be supplied by a simple splash and mechanical separation of the particles when thrown against some hard substance. For low flash-point fuels, such as alcohol or petroleum spirit, used in motor-car engines, this simple method is almost universal. For heavier fuels and those with a comparatively high flash-point, some more elaborate method is necessary, and is generally supplied by spraying the liquid under considerable pressure through a nozzle. By this means the liquid is atomised into minute particles, and in many instances the particles are simultaneously whirled round by centrifugal action, and thus atomisation is increased. In other cases atomisation

is combined with carburetion by using compressed air as the spraying medium; in this way the atomised fuel mingles with the air, which thus becomes more or less carburetted, *i.e.* impregnated with a hydrocarbon.

The completeness of the carburetion not only depends on the degree of the atomisation, but also on the *temperature* of the mixture. The higher the temperature the better the atomisation because the liquid is more easily divided into minute particles, and if the temperature of the mixture is high enough the liquid atoms are vaporised into a gas; in this case the carburetion is practically complete, or nearly so, and good combustion generally ensues. The simplicity of the petrol carburettor has, unfortunately, led many unscientific experimentalists to overlook the essential necessity of complete vaporisation of the fuel before complete carburetion can be obtained, and also the *fact* that complete combustion free from particles of sooty deposit can only be obtained at a temperature sufficiently high to produce complete gaseous carburetion.

The chemical composition of liquid fuels differs somewhat according to their various flash-points and densities, but roughly it may be taken that for complete combustion of 1 lb. of fuel about 15 lb. weight of air are *chemically* necessary. If *volumes* be considered at atmospheric pressure and 62° F., about 200 volumes of air are required for 1 volume of *liquid*; but if the fuel be completely vaporised, then complete combustion may be chemically obtained when the ratio is 6·3 volumes of air to 1 volume of *gas*. Under ordinary working conditions a relative proportion of 10 volumes of air to 1 volume of gas is usually aimed at, and between 8 and 12 volumes produces a fairly satisfactory result; but the proportion varies somewhat for various classes of fuel.

From the above it will be readily seen that the correct mixture of the fuel with air can be best obtained when the fuel is in a gaseous state, in which state the volume at atmospheric pressure and normal temperatures is 30 or more times its volume in the liquid state. The volume of liquid which requires to be admitted to even the largest cylinder at each working stroke is extremely small, and cannot easily be divided or fractionised; but if it is in the state of gas, it can be fractionised with nearly 30 times greater accuracy.

For the highest attainable efficiency it is also evident that the heat required to vaporise the liquid should be obtained from heat

otherwise wasted, such as the exhaust waste or jacket waste, and should not be supplied by the fuel itself inside the cylinder, because this supply reduces the working value of the heat supplied to the engine by reducing its temperature and pressure. It should, however, be pointed out that at the time of writing, in the Diesel engine, which attains a very high thermal efficiency, the fuel is injected as a liquid, but heat is supplied by the high degree of compression.

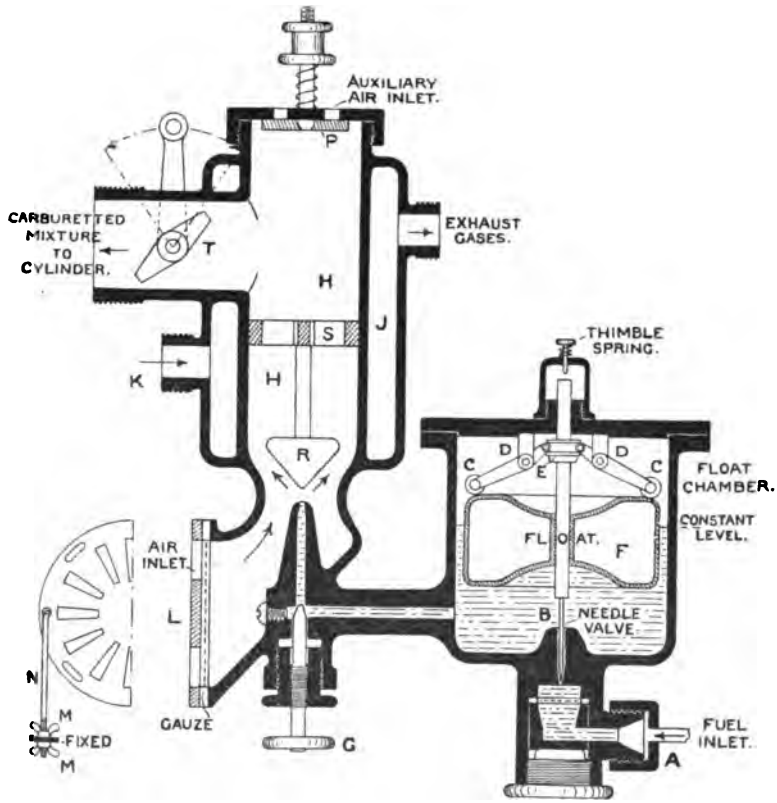


FIG. 382.—Float Feed Carburettor.

Float Feed Carburettor.—For ordinary petrol engines, and for several paraffin and kerosene engines, a carburettor is generally used. Fig. 382 shows the general principle of a well-known type, which combines in itself the three functions of atomisation, vaporisation, and carburetion. The supply of liquid fuel through the

pipe *A*, which is in connection with the fuel tank, is governed by a float connected through levers *CDE*, and a muff coupling *E* with a needle valve *B*. The fuel gravitates into the float chamber, and when the float is raised sufficiently by a change in level of the liquid in the chamber, the float *F* operates the levers *CDE*, fulcrumed at *D*, which push the spindle downwards and close the valve *B*. The supply of fuel to the engine is regulated by the atomiser valve *G*. The vaporising chamber, *H*, is surrounded by a jacket space, *J*, through which a part of the hot exhaust gases are deflected; the proportion of the exhaust gas, and therefore regulation of the temperature, of the vaporiser can usually be effected by a cock fitted near the inlet *K*. The air enters at *L* through a wire-gauze screen, and the supply can be controlled as shown by means of two thumb screws, *M*, on a rod, *N*, in connection with a perforated disc. Additional air can be admitted if required through an auxiliary air inlet valve *P*, which is controlled by a spring and thumb screw. The supply of mixture to the engine can be regulated by a throttle valve, *T*, operated either by hand as required, or by a governor.

The action is somewhat as follows:—When the inlet valve of the engine cylinder is opened and the suction stroke begins, a slight decrease of pressure below atmospheric is created in the vaporising chamber. The difference in pressure draws in a small quantity of liquid through the atomiser valve *G*, which, striking against a roughened cone *R*, is atomised into a fine spray. At the same time air is drawn in through *L*, and mingles intimately with the spray, which is vaporised during its passage through the perforated plate *S* by the heat derived from the jacket *J*. Almost complete carburetion of the air thus takes place within the upper part of the vaporiser, and a sufficient quantity of the mixture passes from the vaporiser to fill the cylinder. When the cylinder inlet valve closes, carburetted air is left in the upper part of the vaporiser, which forms the nucleus of the next charge for the cylinder, and which, because of its higher temperature, assists in producing more complete vaporisation and carburetion. During warm dry weather the hot jacket may not be necessary for carburetting petroleum spirit, but with fuel of higher flash-point, such as paraffin or kerosene, the vaporiser jacket is essential to good carburetion when using this general system. In cold frosty weather the vaporiser, if unjacketed, or if not kept warm by a suitable housing, may become choked by ice or frozen moisture drawn into it with the atmospheric air.

Allsop Vaporiser and Carburettor.—The Allsop system is capable of utilising heavy oils, such as paraffin and kerosene, and claims success with oils of 200° F. flash-point. Three sectional views are shown in Fig. 383 of a two-cylinder engine with a third (pump) cylinder interposed between them and connected with the same crank shaft. Where petrol is to be used for starting, two floats are fitted; and when the engine is warmed through, the petrol float is switched off and the paraffin float substituted. Starting the engine can also be effected by heating the vaporiser by means of a blow-pipe inserted at the plug hole *T*.

The action is as follows:—On the downward stroke of the pump plunger *U*, paraffin and a limited proportion of air enter the vaporiser *V* and the pump cylinder

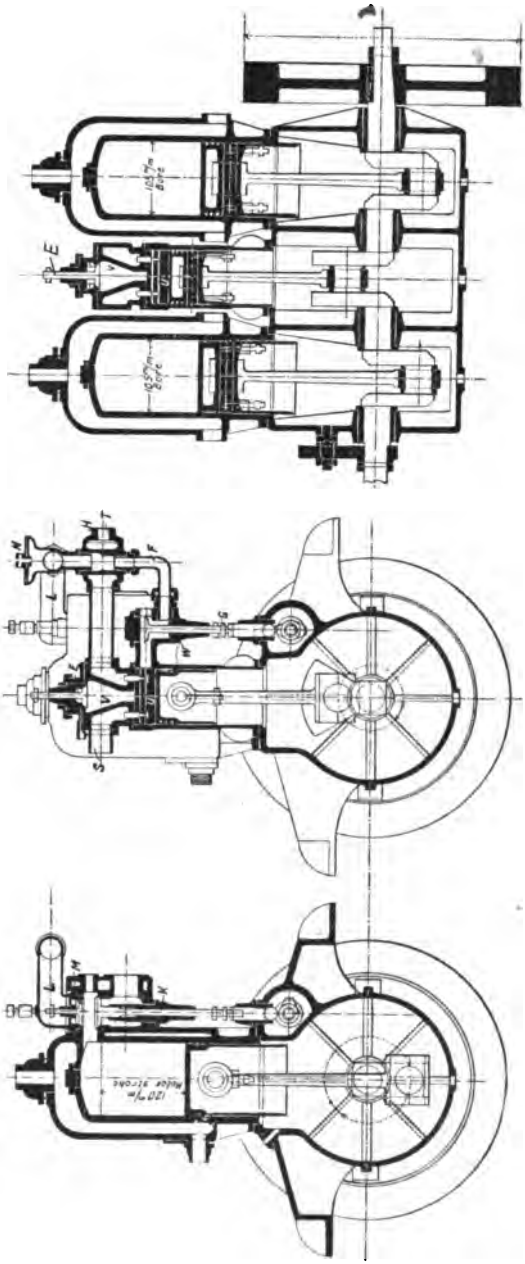


FIG. 383.—Allsop Paraffin Engine.

through the atomising fuel inlet-valve *E*. The vaporiser *V* is heated by the exhaust gases emitted through the mechanically controlled valve *K*, and which pass through the pipe *TS*. On the upward stroke of the plunger *U*, the gaseous mixture is discharged through the controlled valve *W* and pipe *F* to the vapour superheater *H*, where it is kept hot until drawn into either one of the working cylinders through the pipe *L* (fitted with a branch leading into each cylinder) and the automatic inlet *M*. Extra air to ensure complete combustion is drawn in through the screen and air inlet-valve *N*, which is common to the two working cylinders. If required, the air can also be warmed.

It is claimed for the Allsop engine that by varying the amount of air admitted to the pump the amount of fuel drawn in is controlled, and that by the provision of a suitable controlling device between the pump and the working cylinder the amount of charge can be adjusted as easily as for an engine operated by a fixed gas. There is no reason to doubt that the supply of fuel in the gaseous state can be more easily controlled than in the liquid state, and it is, therefore, quite apart from its higher thermal efficiency, the better of the two states in which to supply the fuel to an internal combustion engine. Of the Allsop engine, *The Engineer* (June 21, 1907) says: "We have had an opportunity of witnessing its operation for motor-car work using ordinary paraffin oil, and are able to substantiate the claims made for it as regards completeness of combustion and freedom from visible exhaust fumes under varying conditions."

Ignition.—*Ignition* is obtained in four distinct ways:—

1. *By Contact with a Hot Flame.*—In the early gas engines the application of a constant external flame to a projecting part of the rear end of the cylinder was made to serve as an igniter or firing arrangement. In its simplest form there was a mechanically operated sliding valve with a single slot cut across it; when this slot was brought opposite a similarly formed slot in the standing part of the cylinder, the flame of an ordinary gas jet shot through the opening thus formed into the cylinder and fired the charge, which had been previously admitted near the same place. There was always a chance of back-firing with this method, outwards instead of inwards, and consequently a baffle plate was generally fitted to prevent this happening as far as possible. Flame contact firing is now generally discarded.

2. *By Contact with a Hot Tube.*—The tube is generally made of some refractory material, such as metal or porcelain, and in many cases now two tubes are fitted side by side so that one can be renewed while the other is in use. The tube is heated by an external flame which maintains it at a dull red heat (about 900° F.), except at the instant of firing, when it becomes momentarily more or less incandescent. The charge obtains access to the interior of the tube, which is always in open connection with the interior of the cylinder behind the piston. The instant of ignition is governed by a *timing valve* operated by a cam on the lay, or cam, shaft; but for very small engines the timing valve is usually omitted, and the admission is automatically controlled through a valve which is incapable of time adjustment; or else there is no valve at all, ignition taking place when the compression has augmented the temperature of the tube to that necessary for firing.

3. *By Electrical Ignition*, of which there are several varieties. In the *internal make and break* system, the principle of which is illustrated by Fig. 384, there are two platinum points, *A* and *B*,

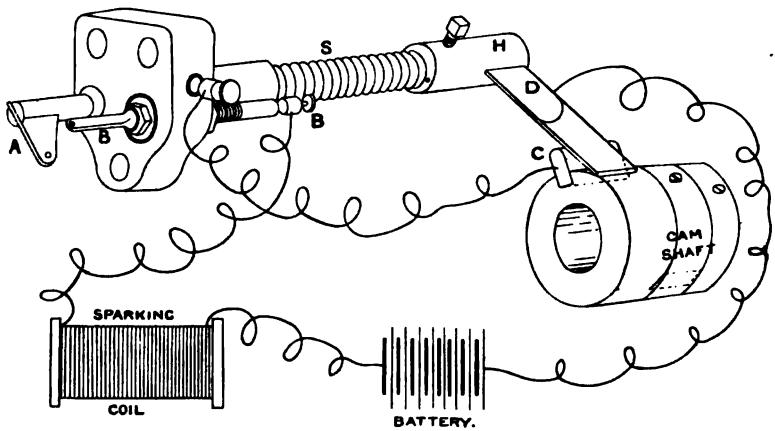


FIG. 384.—Make and Break Electrical Ignition.

inside the cylinder head, and near the admission inlet, which are in connection with a cam *C* and a simple sparking coil and battery. *B* is fixed and insulated from the cylinder. *A* is "earthed" and in electrical contact with the cylinder, and, through the rocking spindle *H*, with the tongue *D*. The spring *S* insures firm contact between the tips *A* and *B* without extreme accuracy of the pro-

portions of the cam, etc., and also gives a sudden separation of the contacts when the cam passes the tip of the tongue *D*. In the position shown no current can pass, but further rotation of the cam joins up *A* and *B*, and a momentary current passes through the circuit, which sets up an inductive current of considerable intensity (but not a high tension current) in the coil. The sudden release of the tongue and parting of the terminals *A* and *B* breaks the circuit, and a fat spark, releasing the current across the terminals *A* and *B*, ignites the charge.

In the *jump-spark* system, shown in principle in Fig. 385, an induction coil is used instead of a simple spark coil, and the terminals *A* and *B* are both stationary, with their tips from 30 to

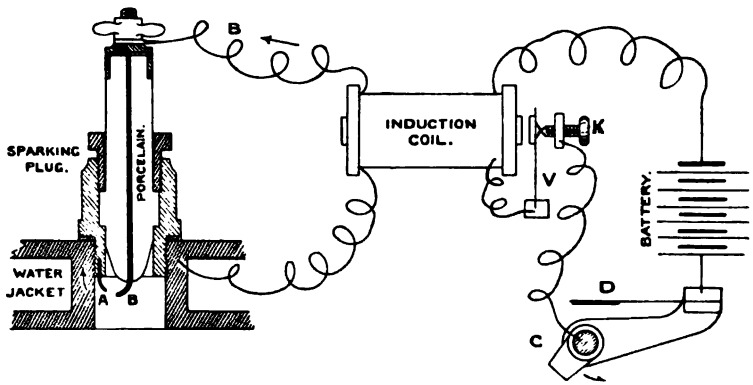


FIG. 385.—Jump-spark Electrical Ignition.

50 thousandths of an inch apart; at 400 volts a spark will jump across an air space up to 30 thousandths of an inch. The points *A* and *B* are insulated from each other by porcelain, or mica, etc., and are fitted in a metal plug screwed into the head of the cylinder. Wires extend from *A* and *B* to the secondary terminals of an induction coil; the circuit in connection with *A* is usually "earthed." The primary winding receives electric current from a battery (or other electrical source), through a circuit-closing cam *C*, a tongue *D*, and a vibrator *V*. *V* consists of a steel spring carrying a soft iron piece opposite the end of the soft iron core of the induction coil. When the primary circuit is open at *CD*, the vibrator spring rests against a contact screw *K*, which closes the circuit at this point. The primary circuit is completed when the cam *C* comes in contact with *D*, and the induction core is then

magnetised, and attracting the vibrator V, opens the circuit at K. A rapid vibration occurs and continues so long as C and D are in contact, and the circuit at K is closed and opened by the successive magnetisation and demagnetisation of the induction core. This action produces a high tension current in the secondary coil of sufficient electromotive force (voltage) to cause a succession of sparks to jump the air gap at the terminals A and B.

A modification, often used, of the above system dispenses with the vibrator V. The secondary circuit being always a closed one, except, of course, for the terminal gap AB, contact is made in the same manner as above, a low tension current flowing through the primary winding when C and D are in contact. At the instant of breaking contact a high tension current is induced in the secondary winding, and a single spark occurs at the gap AB.

It will thus be seen that with this system the charge is fired at the instant of separation of C and D; whilst in the one previously described it is fired immediately after C and D make contact. Simplification of parts and a greater accuracy of timing are claimed for the single-spark system, whilst with the former system it is claimed that the greater heat produced by the series of sparks will fire weaker charges and ensure a greater certainty of firing.

A diagram of the *Simms-Bosch low tension magneto* system of ignition is shown in Fig. 386. The magnet M and armature N are fixed, and a soft iron envelope O is *oscillated* between them by the movement of a connecting rod P worked from a short crank on the end of the cam shaft R. When the envelope O is in the position shown in the right-hand figure the magnets saturate the armature with magnetism, but no current is induced in the armature winding so long as the envelope stands still. The sudden movement of the envelope O into the position shown in the left-hand figure cuts the lines of force between the poles and the armature, which is suddenly demagnetised, and, being instantly remagnetised in the reverse direction, a momentary current is caused in the armature winding of sufficient strength to ignite the mixture. At a certain relative position, generally when the point S becomes separated about $\frac{1}{16}$th inch from the point T, the current is at its maximum tension, and the two sparking points (of the type shown in Fig. 384) in the cylinder should be separated at the same instant to produce a spark and ignition. By an ingenious arrange-

ment the cam can be moved axially along its shaft on a spiral feather, and the *timing* of ignition can thus be altered within the necessary range.

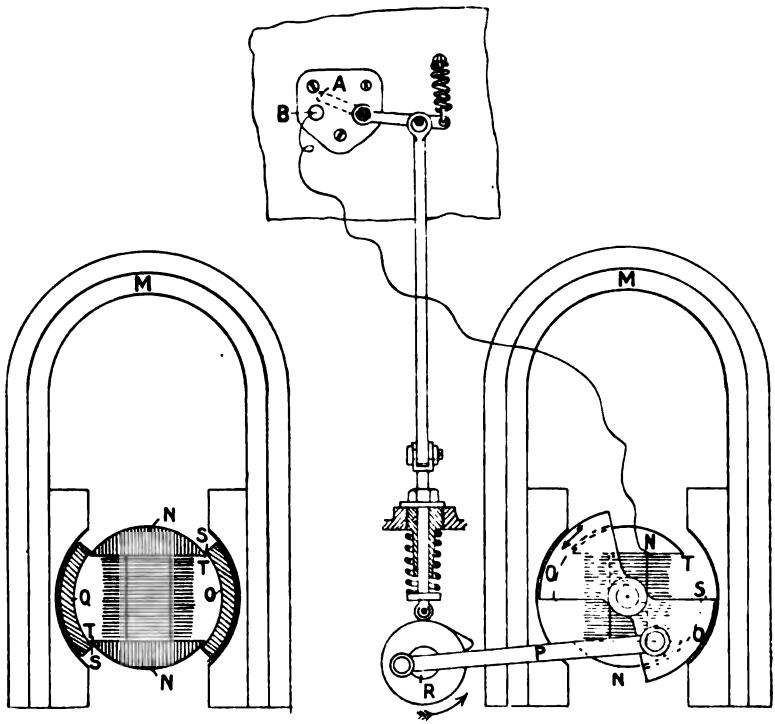


FIG. 386.—Simms-Bosch Low Tension Magneto-Ignition.

In the Simms high tension system the armature revolves instead of oscillating, and a sparking plug of the usual type is used, being connected with a secondary winding as in the jump-spark system previously described.

The general principle of a *high tension* (*dynamo*) ignition system is shown in Fig. 387, in conjunction with a Pognon sparking plug. This plug has a space left around the point *B*, so that when ignition takes place a certain amount of expansion is obtained within this space, and it is claimed that the gases, in rushing past the points, tend to clean them. The armature is revolved (not oscillated) either directly by the shaft or through gearing, and is compound wound with both primary and secondary

coils. The commutator contacts are not shown in the sketch, but can easily be traced when the actual arrangement is seen. The secondary wire is, when working, permanently connected with one of the sparking points, but the primary and thicker wire is connected through a condenser and a contact breaker. The armature and commutator form essential parts of a small dynamo, giving a very high voltage (or tension) with, however, only a very small current strength (ampères), so that the electrical power is actually very small. The condenser acts as a safety valve to relieve extreme pressure or tension, which may occur under certain working conditions, and thus prevents the possible fusing of the secondary-wire connections. High tension ignition is very reliable,

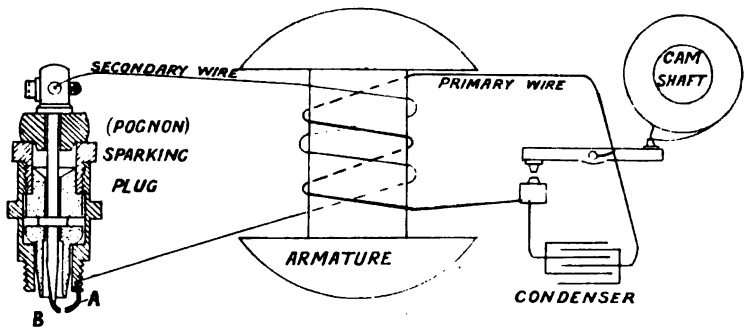


FIG. 387.—High Tension Dynamo-Ignition.

because the discharge jumps across the sparking points easily even when they are dirty, but a very good insulation of the plug is necessary to resist the high voltage and possible leakage. Mica is generally used instead of porcelain for the plug insulation, but several makers still prefer to use porcelain of special quality.

4. *By Compression.*—When air is compressed the energy expended in compressing it is stored up in the air in the form of heat, but in practice there is, of course, some loss from radiation, etc. The increased temperature of the air due to the increased heat stored up in it is accompanied by a corresponding increase of pressure, and if no loss took place a temperature given very approximately by the formula below might be obtained—

$$\frac{T_0}{t_0} = \left(\frac{P}{p}\right)^{\frac{2}{7}}; \text{ or, } \log T_0 = \log t_0 + \frac{2}{7} \log \frac{P}{p};$$

where T_0 is the final absolute temperature, and P the corresponding pressure;

and t_0 is the initial absolute temperature, and p the corresponding pressure.

But in actual practice only about one-half the rise thus calculated takes place, because it is necessary to keep the cylinder at a moderately safe temperature by cooling it, as already explained, and an average value is given by the equation—

$$\log T_0 = \log t_0 + \left(\frac{1}{4} \text{ to } \frac{2}{9}\right) \log \frac{P}{p}.$$

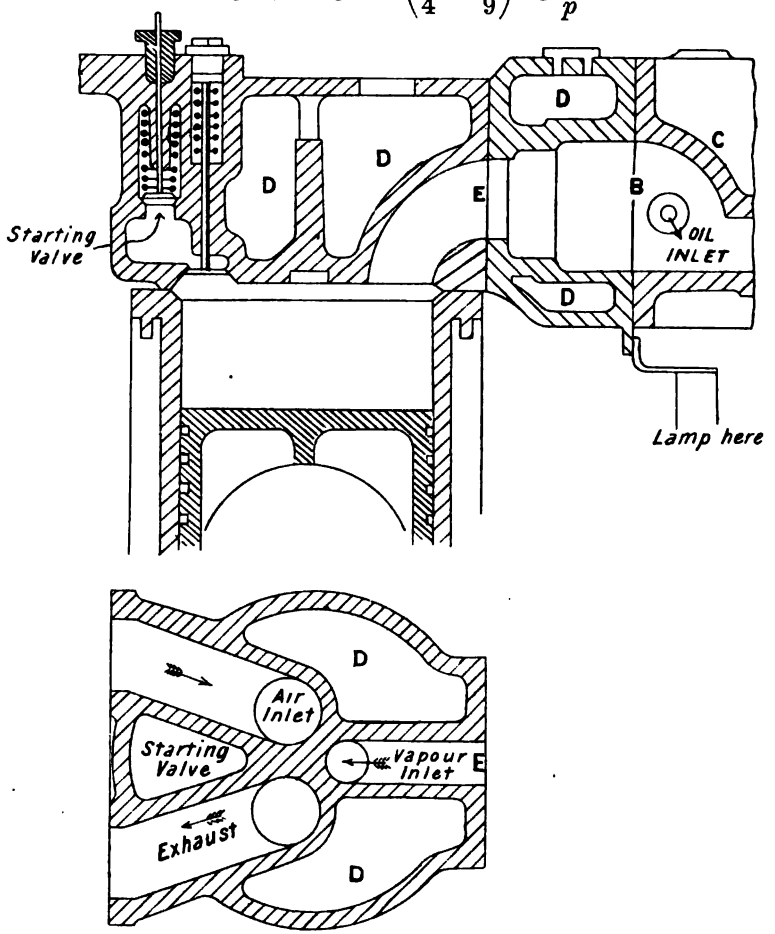


FIG. 388.—Hornsby Vaporiser.

(*a*) In the *Hornsby engine* the heat generated by compression

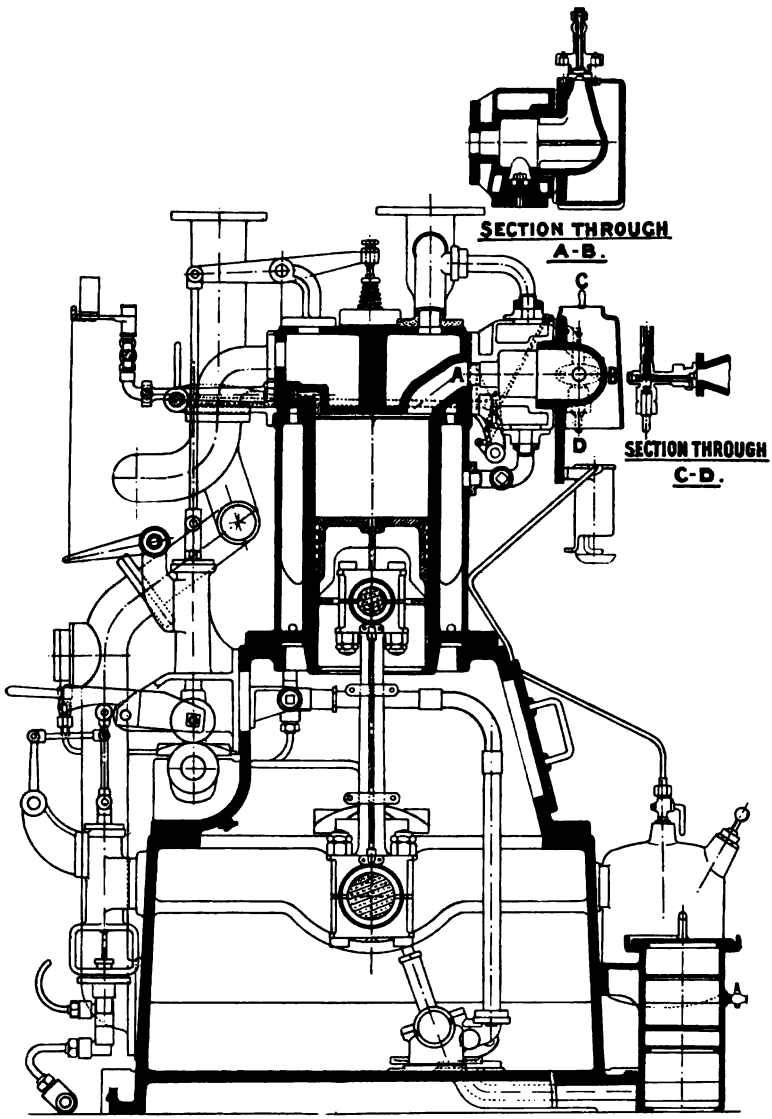


FIG. 388A.—Cross-Section of Hornsby Oil Engine.

alone is insufficient to produce the necessary temperature for firing heavy oils, so that although such compression materially contributes

towards the desired result, the oil charge is injected at the correct

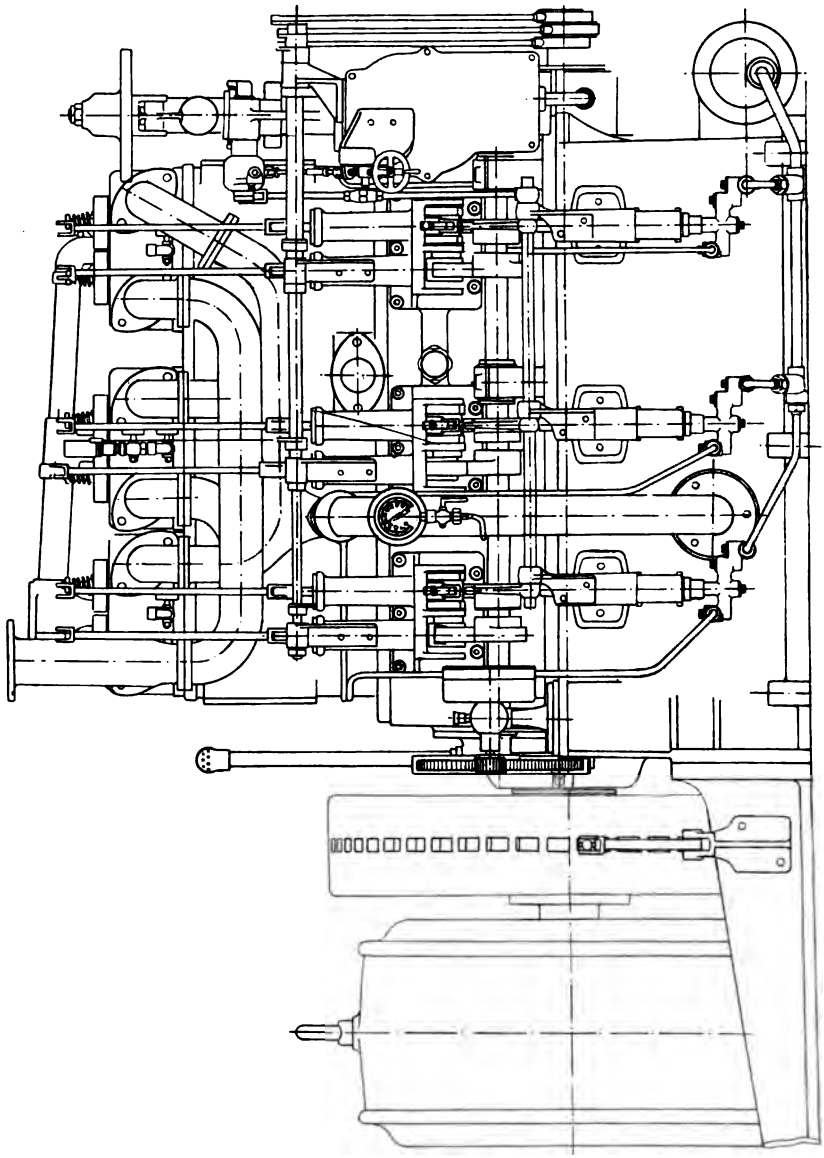


FIG. 389.—Front Elevation of Hornsby Oil Engine.

instant on to a vaporiser. The Hornsby vaporiser *C*, Fig. 388, as fitted to a vertical engine, forms part of a chamber *B* on the side

of the cover of the engine cylinder with which it is connected. Before starting, the vaporiser is warmed by applying some heating apparatus of the blow-pipe type. The vaporiser retains sufficient temperature after the engine is started—in conjunction with the temperature caused by the compression—to cause combustion of

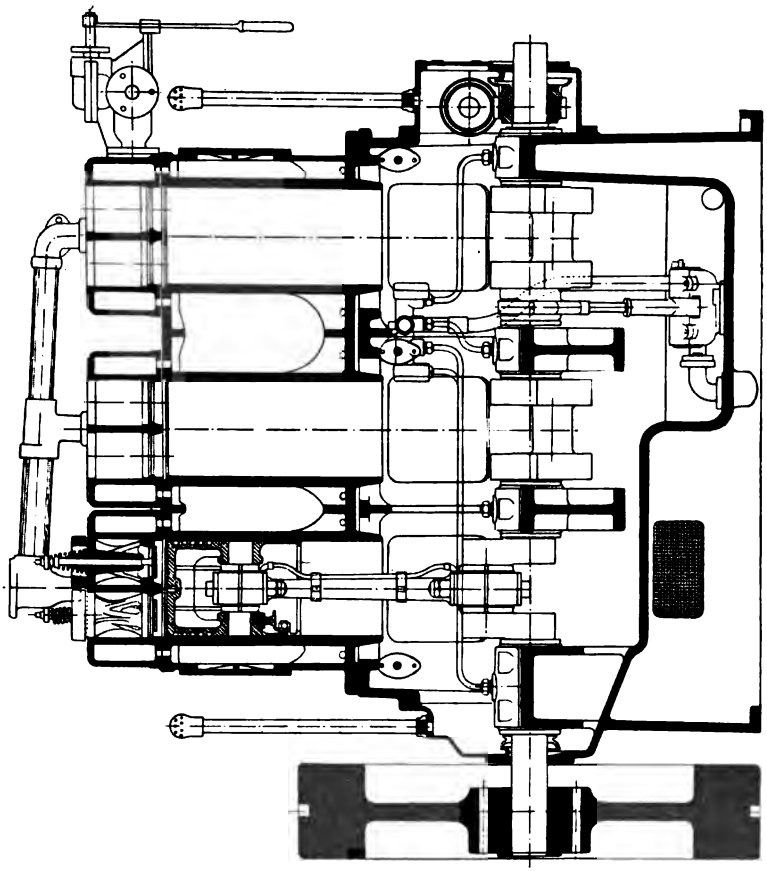


FIG. 390.—Sectional Elevation of Hornsby Oil Engine.

the next charge, which is injected into it in the form of spray. The vaporiser *C*, although connected with an extension of the cylinder cover, is not water jacketed, and is simply air cooled. The remaining part of the chamber, that next to the cylinder, is water jacketed, and its temperature can be regulated by controlling the supply of cooling water to *D* through a cock on the inlet pipe below. Large engines are set in motion by compressed

air stored in a reservoir or pumped by hand, and admitted through the starting valve.

Figs. 388A, 389, and 390 show various views of the construction and arrangement of a 3-cylinder Hornsby oil engine, as fitted in several warships for using ordinary liquid fuel. Among other details, these figures show: the vaporiser with the lamp-bracket and blow-pipe apparatus used for starting; the indicator and connections for taking diagrams from each cylinder; the separate fuel pumps, the stroke of which is regulated by the governor; and the forced lubrication pump and pipe connections. In recent types, a cupped ring is fitted below each working barrel to collect the surplus lubricating oil and, by means of a drain-pipe, to return it to the pump suction.

(*b*) In the *Diesel engine*, shown in Figs. 392 to 402, ignition is obtained by high compression alone. The clearance space behind the piston is about 7 per cent of the volume struck out by the piston, and thus during the compression stroke 107 volumes of air are compressed into 7 volumes. If no rise in temperature took place the pressure would then be about 15 atmospheres, or 220 lb. per square inch; but a rise in temperature must and does take place: firstly, because the air is warmed by contact with the cylinder metal; and, secondly, because the work done in compressing the air is stored up in it in the form of heat. Thus, at the completion of the compression stroke the air is compressed to about 35 atmospheres, or 500 lb. per square inch, and attains a temperature of about 1000° to 1450° F. The compression is neither isothermal nor adiabatic, but generally conforms more closely to the latter, and, $P.V^{1\cdot3} = \text{constant}$, gives about an average value.

Cooling.—It is necessary in all internal combustion engines to cool the cylinder, the cylinder end, and the valve casings which are generally attached to it, for two reasons: (*a*) to prevent overheating of the metal which, when the temperature is increased above about 600° F., is rapidly reduced in its strength; (*b*) to allow efficient lubrication, though with oils now obtainable having a flash-point of 550° to 600° F. it is not so difficult of attainment as in former times. For the same reasons it is necessary, in large engines running at high speeds, to cool the piston and rod. The upper limit of temperature allowable for the metal of the internal working parts is thus about 550° F., and even this comparatively moderate limit is seldom reached.

It is also necessary to make the piston and its packing rings fit fairly accurately in the working barrel, to prevent leakage. With the high temperatures obtained from the explosive gases, the piston, being unable to radiate its heat so easily as the cylinder, expands in a greater degree, and therefore becomes too tight a fit in the cylinder to obtain a high mechanical efficiency. The objection to cooling is that heat is abstracted from the cylinder, and, in its turn, from the heat energy contained in the gases produced by combustion; consequently the lower the limit of temperature to which the cylinder is cooled the greater is the loss of heat.[1] The "heat rejected" by cooling amounts to about 30 to 45 per cent for cylinders of 20 H.P., and for larger cylinders is less, possibly only 20 per cent of the total heat energy contained in the fuel.

For small motors, up to three- or five-horse power, such as used for cycles and some small boats, *air cooling* is sometimes adopted. The cylinder body (and end, in many cases) is made with a number of thin, deep ribs or rings, projecting, generally radially, from the body and from the end. These ribs being made in one piece with either the cylinder or its end, rapidly radiate heat away to the surrounding atmosphere. With motor cycles, rapidity of motion through the air increases the cooling action; and in cases where the motion is insufficiently rapid for this purpose, a fan, driven by the motor, is used either to impel air on to or to draw it past the cylinder. An example of insufficiency of air cooling is when a motor cycle is climbing a hill and its maximum power is required; great heat is given out by the engine, which the lack of draught from the slow motion of the cycle fails to convey away, and the cylinder warms to a temperature approaching a dull red heat of about 900° F. Air cooling has also been adopted for motors of much larger horse-power, and for these a large fan power is necessary; some considerable saving in weight is made in this way over the water cooling next described. In cold climates air cooling is preferable to water cooling, because the trouble of draining the water jackets to prevent fracture by freezing when the motor is not in use is avoided. The saving in weight is, however, generally effected at the cost of efficiency, because a considerable fraction of the power developed by the engine is absorbed in driving the fan.

Water cooling is generally used for engines of over three horse-

[1] See also remarks later on *Suction stroke*, where the advantage of a low temperature points to an increase of *power*, but not to an economy of *fuel*.

power. The cylinder body or end, or both, are made with jacket spaces through which water is circulated. The cooling outfit, in addition to the jacket spaces, generally consists of three parts: a tank, a radiator, and a pump; and, occasionally, a filter or oil separator. The pump is usually fitted in the coolest part of the system of circulation, which is just before the water enters the jacket spaces; it should be of a type either centrifugal or rotary, or preferably helical, which does not materially retard the flow of circulation should it fail for any reason.

For locomotive work, where the quantity of water is necessarily limited, the same water is used over and over again by cooling it during circulation through a radiator and tank fitted in series with the cylinder jacket. Radiators vary in design: a simple type, still to be seen in many motor cars, consists of a number of gilled tubes (or of one continuous gilled tube folded backwards and forwards over itself) placed in the most exposed position, so as to meet with the greatest possible air draught when the car is in motion. The hot water from the jackets passes into the radiator, where it gives up its heat to the metal of the tubes, which in turn radiate it to the atmosphere. A fan is often fitted in conjunction with the radiator, partly to assist the draught when the car is moving, but principally to cause one when the car is at rest with the engine running. With stationary engines, if the supply of water be limited and a continuous stream is not available, the tank is made large enough to act as its own heat radiator, and neither gilled pipes nor fans are necessary.

For marine work the tank is usually unnecessary, and a continuous supply of cooling water can be obtained from outside the vessel; but for continuous sea-work special arrangements must be made for removing the lime scale deposited in the jacket when sea-water is used and high jacket temperatures are maintained. A circulating pump is frequently unnecessary, and in well-arranged water-cooling systems natural circulation is often relied on. Perhaps the best example of natural circulation is afforded by that of a properly designed kitchen boiler hot-water plant with its pipe and tank connections.

With a vertical inverted cylinder the natural direction of circulation is upwards from the lower end of the cylinder body jacket into the end jacket, and then to the upper part of the radiator, where the cooling assists the downward circulation to the

tank and pump. This direction of circulation is generally adopted for the Diesel engine (see Fig. 392), and is a suitable one because the ignition or firing is dependent on the temperature produced by compression alone; the hotter water is near the point of ignition. Any excessive cooling action (which might possibly be produced if the direction of circulation were reversed) tends to reduce the temperature obtained by compression and so prevent proper ignition. The colder circulating water, surrounding the lower end of the cylinder, tends to reduce the temperature, and therefore the pressure during the working stroke. The adverse effect produced by this cooling action is, however, in engines where the charge is ignited by compression, counteracted by the comparative slowness of the combustion which, by what is known as "after burning," tends to maintain, and does in practice generally maintain, the pressure in the cylinder above that obtained by a more nearly instantaneous combustion, notwithstanding the comparatively more rapid cooling towards the end of the working stroke.

In other engines, looking at the problem from a general point of view, where an independent form of ignition, such as electrical, is used, it appears preferable to adopt a circulation in the opposite direction, *i.e.* first through the end jacket, and then through the body jacket from back to front (or top to bottom). In this way a lower maximum pressure and a greater mean pressure may be obtained, because the cooling effect of the water jacket may cause slower combustion after firing, while the higher temperature of the jacket water near the end of the working stroke has less effect in reducing the temperature during the stroke. With a vertical cylinder the reversal of the natural direction of flow means that a pump must be employed, but with a horizontal cylinder natural direction of circulation can be followed.

The tank can be placed as convenient, but as it is important that the water jackets and the radiator should be completely filled, the level of the water in the tank should be above the highest points which require to be filled. For this purpose a vertical pipe with an open mouth is frequently fitted to the tank, and serves also as a filling pipe; air-cocks or plugs, as convenient, are fitted at the highest points in the circuit to allow any air to escape when filling. In some instances a tank is non-essential, and its place and function, which is simply to automatically prevent overheating and

to make up loss by evaporation, is taken by the vertical filling pipe, the open mouth of which prevents an accumulation of pressure.

The usual temperature of the water on leaving the jackets is about 170° to 180° F., but in some instances higher temperatures are allowed and some steam is formed; a pressure above atmospheric is seldom permitted, although there is a growing tendency to utilise the water jacket as a boiler to supply steam under pressure for atomising and vaporising the oil on entering the cylinder. It is claimed for this system that an increase of efficiency can be obtained, because heat, otherwise rejected to the cooling water, is returned to the engine and does work on the piston. Without the results of practical experiment, any forecast as to the possible increase or decrease of efficiency due to this method is entirely speculative. Assuming the mean temperature of the jacket water to be about 150° F., and the mean temperature of the products of combustion near the working barrel to be about 550° F., the mean temperature of the cylinder metal is probably about 350° F., which is a temperature which could be increased without danger to the metal and with some increase of working pressure during expansion.

Suction Stroke.—To get the maximum *power* from any internal combustion engine it is necessary, during each cycle of operations, to introduce the greatest possible weight of combustible materials, provided always that they are in their correct proportion for complete chemical combustion. A fixed proportion of oxygen is necessary to complete combustion of the fuel, which by weight is 1 of fuel to about 8 to 12 of air, according to the fuel used. The ratio for any particular fuel may be generally found by direct experiment, or by calculation when the composition of the fuel is fairly accurately known. The greatest weight of any given fuel can be burnt when the greatest weight of air is admitted for its combustion. Certain causes, more or less dependent on each other, tend to reduce this quantity, viz.—

(*a*) The flow of air into the cylinder must be caused by difference of pressure, and therefore the pressure inside the cylinder must be less than that outside it, *i.e.* the atmosphere generally. For this reason a mechanically opened valve is preferable, as the difference in pressure is then reduced.

(*b*) The pressure of the atmosphere varies with conditions of weather, height above sea-level, etc. At great heights, the power of the engine is consequently reduced.

(*c*) The gases in the clearance space, having a pressure slightly above atmospheric when the inlet is opened, expand and resist the inflow of air.

(*d*) As the air enters, and comes in contact with the hot cylinder walls and piston, already warmed by previous firings, it becomes heated and expands, and therefore less weight can enter. If the air is supplied directly from the surrounding atmosphere, the pressure therefore being almost atmospheric and practically constant, the greatest weight of air enters the cylinder when the difference of absolute temperature of the gas or air inside and outside the cylinder is least.

(*e*) The admission valve can be neither closed nor opened instantaneously, both operations are more or less gradual, and it takes an appreciable time for the air to enter.

(*f*) During admission the piston may have completed part of the return stroke, and consequently the cylinder is only partially filled with air at some pressure which must be less than that of the supply.

From the foregoing reasons it may be concluded, if the greatest possible *power* is to be developed in the cylinder, that the cylinder, piston, etc., with which the entering air comes in contact should be as cool as possible. This conclusion is not consistent with the greatest possible *economy* in practice, and a compromise is arranged. (See also remarks on *cooling*.)

Compression Stroke.—Both during the admission and after the air inlet valve is closed, heat passes into the charge from the cylinder walls, etc.; the pressure rises, and therefore the work done during compression is increased. This work of compression is done by the fly-wheel, or other external agency, at the expense of energy previously accumulated in the mechanism, and therefore increases the wasteful expenditure of energy during compression. Evidently, if the original temperature is high, the final temperature becomes very much higher; and the liability to premature ignition, which frequently occurs for this reason, is increased. On the other hand, after the temperature is raised, by compression, above that of the cylinder walls, heat is given out to the walls (energy is thus wasted), but there is a tendency to decrease the temperature otherwise due to compression in the later part of the stroke. Ignition is generally made to occur before the end of the compression stroke, but the exact position depends on the nature of the fuel and the speed of

the piston; the fuel, it might be again stated, is usually already in a gaseous state at the instant of ignition. Ignition should be so timed that the maximum temperature is attained before the piston has completed one-twentieth of its working stroke, but it should not be early enough to exert any appreciable retarding effect on the turning motion of the shaft. (In certain engines, such as the Ailsa Craig type, where the crank axis is not in the same plane as the cylinder axis, the ignition is usually arranged to occur later than in the ordinary type.)

Working Stroke.—As the piston begins to move forward, combustion continues and tends to maintain the highest pressure attained by constantly adding heat; but the movement of the piston rapidly increases the volume, and consequently the pressure falls after about one-tenth of the stroke is completed. In many cases the fall of pressure begins with the stroke, and is due to the increase in volume not being compensated by increase in temperature. It should not be forgotten that at the very high temperatures which are easily produced in internal combustion engines, carbonic acid gas, or carbonic anhydride (CO_2), and water vapour or steam (H_2O), become dissociated respectively into their chemical elements, and heat is taken from the gases to effect these dissociations; temperature and pressure therefore tend to fall for this reason. In some later position, when the temperature has dropped considerably, these elements, very possibly, again become chemically united by combustion; exactly the same quantity of heat is then returned to the gases as was previously extracted to effect their dissociation. The above is probably the reason for the apparent addition of heat during the working stroke, which is a matter of common observation during experiments, though it may also be due to "after-burning" and slow combustion.

After-burning is more or less common with all types of gas and oil engines, and is a continuation of combustion after the first instant of ignition or firing. It tends to maintain the temperature and pressure, so that the actual expansion curve, as shown by the indicator diagram, is more nearly adiabatic, and in some cases above the theoretically adiabatic curve. If after-burning is too prolonged, or if the stroke is too short, some portion of the fuel passes away unconsumed with the exhaust products, and thus occasions a loss; it is generally caused by too great an admixture of air, the mixture then becoming too weak in fuel.

Scavenging Stroke and Exhaust Control.—The exhaust valve, if one is fitted (it is unnecessary in the ordinary two-stroke cycle engine), is nearly always operated mechanically by a cam on the lay shaft. The instants of opening and closing are often capable of adjustment while the engine is working.

With exhaust control, as with ignition and admission of the charge, the element of time is of first importance to efficiency. There are now many useful data which show the time necessary for the several operations. Only a general statement can here be made, but recent experiments should be studied in relation to the speeds at which the crank and piston are moving respectively.

The exhaust should be opened as late as possible to avoid the too early escape of the exhaust products, which may still have some energy left that has not been used up in doing work on the piston; in certain cases a retardation of the exhaust opening has resulted in a gain of efficiency and improvement in power. At the dead points the piston moves relatively only a small distance while the crank travels through a comparatively large angle; it is near this point, therefore, that the necessary time required for exhausting can be best given without losing either power, which might possibly be otherwise developed in the working stroke, or compression, which is only very slightly diminished.

Engines with a short stroke and great speed of revolution are liable to great loss by exhaust waste; that is to say, by the escape of the exhaust before all possible work has been obtained from it. With high piston speeds, say about 700 feet per minute, the exhaust valve is generally opened at a point between 8 and 9 tenths of the working stroke and closed just after the beginning of the suction stroke. With moderate piston speeds, say about 500 feet per minute, the instant of exhaust opening is later, and the closing can be effected at the beginning of the suction stroke. The piston speed of modern engines is now being increased, following on the lines of the best naval practice, and 800 feet per minute is becoming usual for many large engines.

The effect of steam or *water injection* during the combustion is shown by the indicator diagram in Fig. 391. The full lines show the ordinary diagram obtained without steam or water, and the dotted lines the effect of introducing a slight quantity of water, which is almost instantly converted into superheated steam of about 150 lb. pressure. Instead of a high peak of pressure, there is a

tendency towards a more uniform pressure throughout the working stroke; the temperature in the cylinder is probably less, and, if too great a quantity of water or steam is introduced, the combustion may therefore be less complete; there is about the same mean pressure, but on account of the lowering of the maximum pressure the cylinder need not be so strong, and a considerable saving in weight might be effected if the system could be relied upon; and in consequence of the more uniform pressure during the working stroke the turning moment, or torque, on the shaft is more uniform. This system also commends itself as suitable for a double-acting engine, because the lower pressure and temperature reduce the

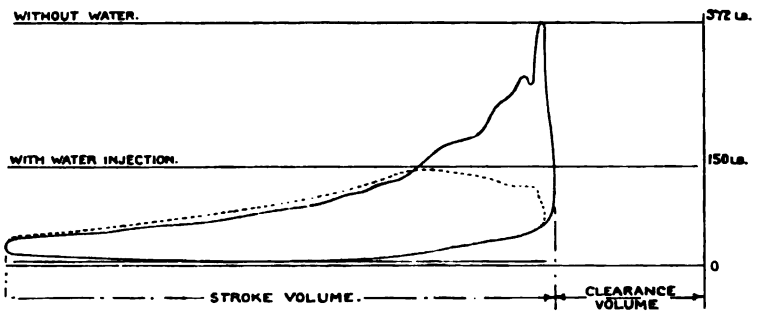


FIG. 391.—Effect of Water Injection.

mechanical troubles of gland-packing, lubrication, and piston and rod cooling.

Back-Firing.—Pre-ignition caused by incorrect timing of the ignition apparatus produces a too early explosion in the cylinder which is technically known as back-firing or firing back. This results in a tendency to reverse the direction of motion of the shaft by preventing the completion of the compression, or upward, stroke. Even if reversal is not effected on the first occasion of firing back, it is likely to occur during the next compression stroke or strokes. In many engines working on the two-stroke cycle, pre-ignition obtained by alteration of the timing is used to effect reversal of direction of motion of the shaft. In other engines, particularly of the Hornsby vaporiser type, although no alteration in timing is made, reversal is caused automatically after the engine is thoroughly warmed and the load is removed or considerably decreased. Back-firing occurs in the cylinder, but other troubles are caused by external firing which are enumerated below.

External Firing.—Troubles of this kind are usually attributable to :—

1. *Short circuiting* of the sparking apparatus, which produces ignition at the wrong instant.

2. *Bad ignition* from weak electric current, dirty sparking plugs, or imperfect mixture or proportion of the combustible and air. In both cases a charge of gas and air is expelled into the exhaust pipe without being properly fired, and this is ignited when the exhaust valve is opened by the incandescent products of the next charge which is ignited. Ignition and firing occurring in the exhaust passage and pipe is technically termed *exhaust explosion*, and is particularly liable to occur after a miss-fire.

3. *Derangement of the valves*, when hot-tube ignition is used and the charge is fired before the inlet valve is closed; but if the valve is already closed and ignition occurs too early, the trouble is an internal one and is actually back-firing, as pointed out above.

4. *Leaky inlet valve* allowing the charge in the cylinder to communicate with the charge outside the cylinder. Firing a charge in the mixing chamber is sometimes attributable to slow combustion and a flame lingering in the cylinder until the inlet valve again opens. In a very bad case the flame may reach the petrol or gas-pipe conveying the charge to the carburettor, and for this reason gauze discs are usually inserted across the induction pipe to minimise the risk.

5. *Over-lubrication* may produce, when comparatively low flash-point oils are used for the purpose, a too rich mixture from vaporisation of the lubricant. Flame lingers in the cylinder and, when the inlet is opened, fires the charge prematurely. Over-lubrication, however, is usually the cause of considerable waste, and in many cases the unpleasant smell of the exhaust is attributable to it as well as to imperfect combustion of the charge. Also gritty carbon particles are left on walls of the cylinder and piston which lead to excessive wear and friction, and in a very bad case may cause the piston to stick in the cylinder.

Diesel Engine.—The engine, Figs. 392 and 393, works on the four-stroke cycle and is usually vertical. The upper part of the frame forms a water jacket A which encloses a cylinder B of close-grained cast iron. The cylinder cover is deep and hollow, and the space C forms a jacket into which the cooling water enters through

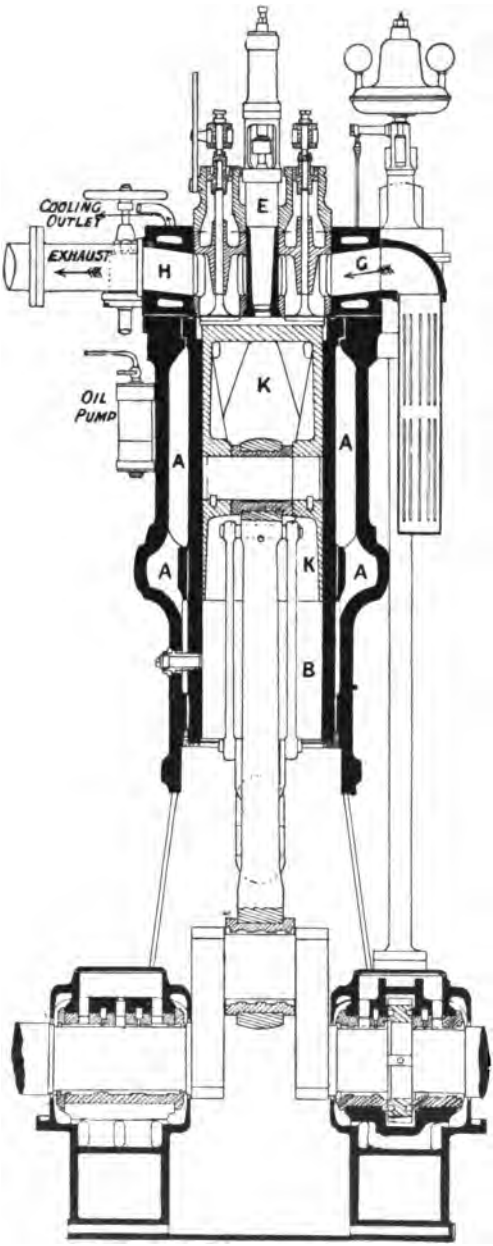


FIG. 392.—Diesel Four-stroke Engine.

D from the cylinder jacket space. Near the middle of the cover is fitted the oil sprayer *E* (for detail see Fig. 399), and near it is a starting valve *F* through which compressed air from a storage reservoir can be passed. On the right is the air inlet *G*, and on the left the exhaust *H*. All the valves are spring-loaded. Both the air inlet and the exhaust outlet are kept closed by pressure in the cylinder, and are opened by bent rocking levers actuated by the cam shaft *T*. The air-compressing pump *L* is water-jacketed; just before the end of the engine's compression stroke, air from the cylinder enters the pump, which, still further compressing the air, delivers it to the air-blast reservoir. From the reservoir the air blast is used to inject petroleum, at 49 to 55 atmospheres pressure, into the cylinder. Any surplus air or air pressure is conveyed to a starting reservoir. The makers state that the Diesel engine is always as ready and as easy to start as a steam engine; air at about 600 lb. pressure can be admitted to the cylinder

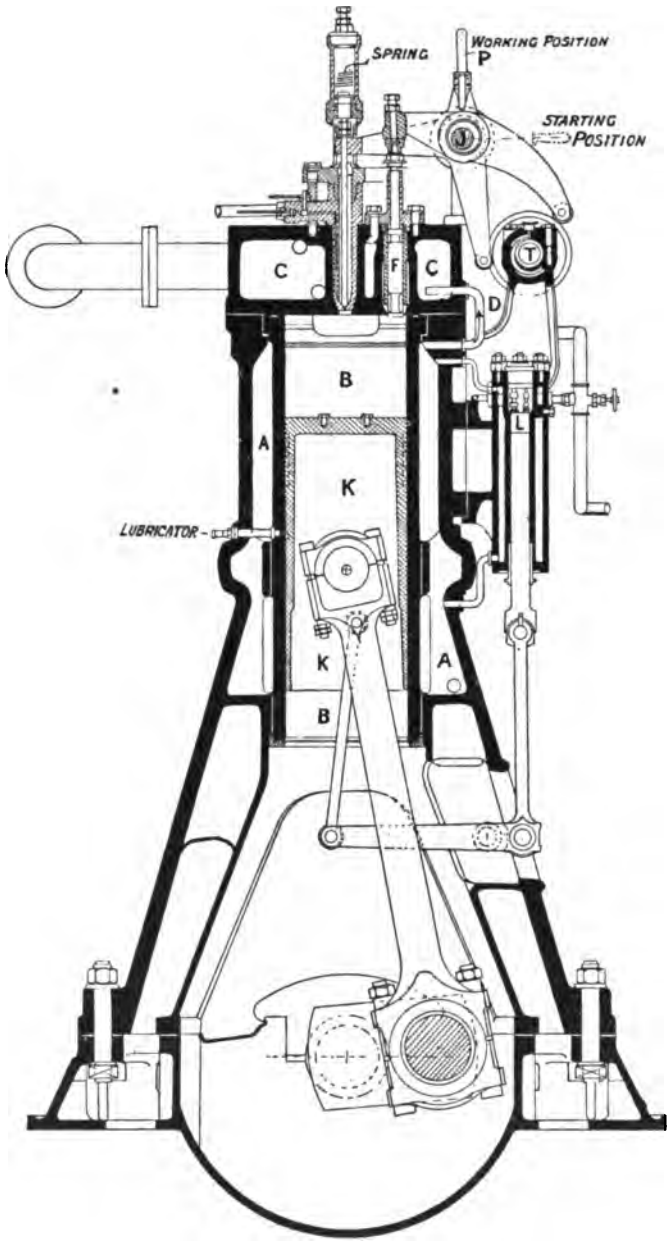


FIG. 393.—Diesel Four-stroke Engine.

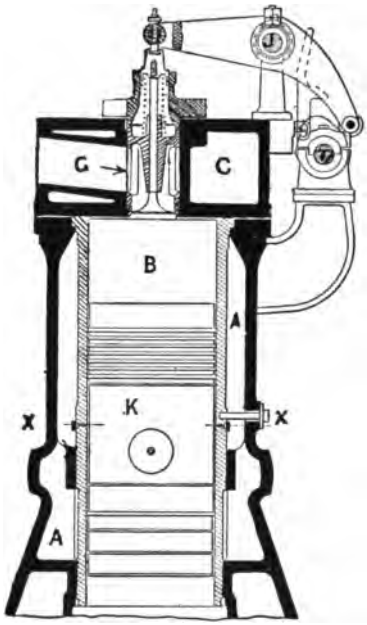


FIG. 394.—Air Inlet Valve.

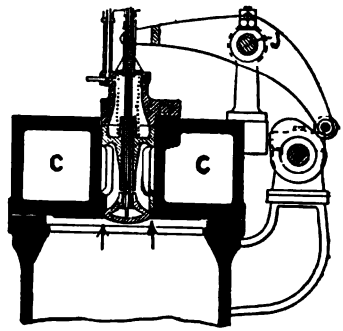


FIG. 395.—Exhaust Valve.

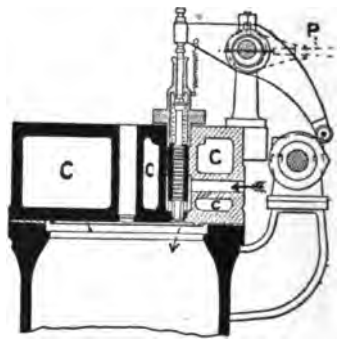


FIG. 396.—Starting Valve.

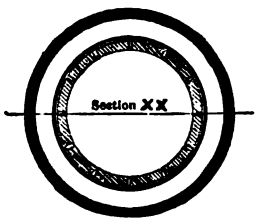


FIG. 398.—Lubrication Holes and Channel.

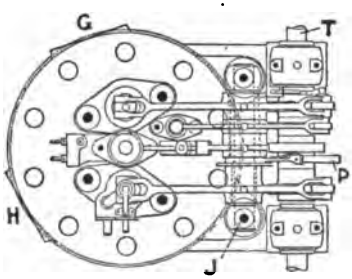


FIG. 397.—Rocking Levers and Starting Lever.

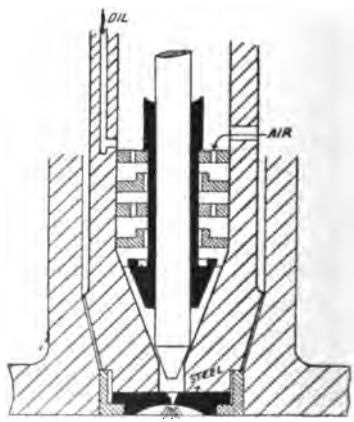


FIG. 399.—Oil Fuel Valve and Atomiser.

FIGS. 394 to 399.—Diesel Valve Mechanisms and Lubrication.

through *F* for starting purposes, and with a proper arrangement of cylinders and cranks the engine can thus be started in any position.

In Figs. 394 to 399, a series of sections and a plan of the head of the cylinder is shown with the various cams and valve mechanisms, as fitted by Messrs. Carels frères of Ghent. Fig. 394 shows the air-inlet valve, which is operated by the rocking lever shown and a cam on the cam shaft *T*. The inlet valve is not water-cooled, but the exhaust valve, which is operated by a similar mechanism, as shown in Fig. 395, is water-cooled and hollow. Water flows downward into the valve cavity through a tube fitted inside a hollow spindle, and upward through the space between the tube and hole in the spindle, to a stationary overflow at the top end. The air-starting valve *F* is shown in Fig. 396, and, its cam mechanism being fitted alongside the fuel inlet mechanism, can be thrown out of gear by a lever *P* shown in dotted lines for the starting position and in full lines in Fig. 393 for the working position. This lever rotates the shaft *J* and eccentric fulcrums, on which the fuel inlet and starting rocking levers pivot, in such a way that the starting valve is closed when the fuel valve is in action, and *vice versa*. The four rocking levers are shown in plan in Fig. 397, which, reading from the top of the illustration, occur in the following order: air inlet, starting air inlet, fuel valve with pipe connections for air and fuel respectively, and exhaust outlet with two pipe connections for inlet and outlet cooling water.

Fig. 398 shows the system of cylinder lubrication. A series of holes is drilled into the inner or working (liner) barrel of the cylinder, at the position shown in Fig. 393 and *XX* in Fig. 394; a groove is then cut circumferentially on the exterior of the liner, and over this groove a ring is shrunk on so as to form a channel in communication with all the holes. A connection is formed between this channel and the outside of the cylinder, and oil is forced in by a pump when the piston is on the bottom dead centre and the holes are opposite the space between the bottom piston ring and the one next above it.

Fig. 399 shows an enlarged section of the Diesel oil fuel valve and atomiser. The oil enters under pressure from the oil fuel pump through the channel shown, and is atomised by an air blast of 750 to 800 lb. pressure per square inch in passing through several (usually four) perforated plates. The holes are about eight-hundredths of an inch in diameter, and zigzagged out of line. The

atomised fuel and air pass through a steel nozzle, and meeting the air in the cylinder, heated to a temperature of about 1000° F. by compression, ignition ensues, and the combustion proceeds steadily and comparatively slowly for an appreciable part of the stroke, as shown by the indicator diagrams in Fig. 401. In this system it should be noticed that the oil enters the cylinder as a liquid, not as a vapour. Vaporisation takes place almost instantaneously, but

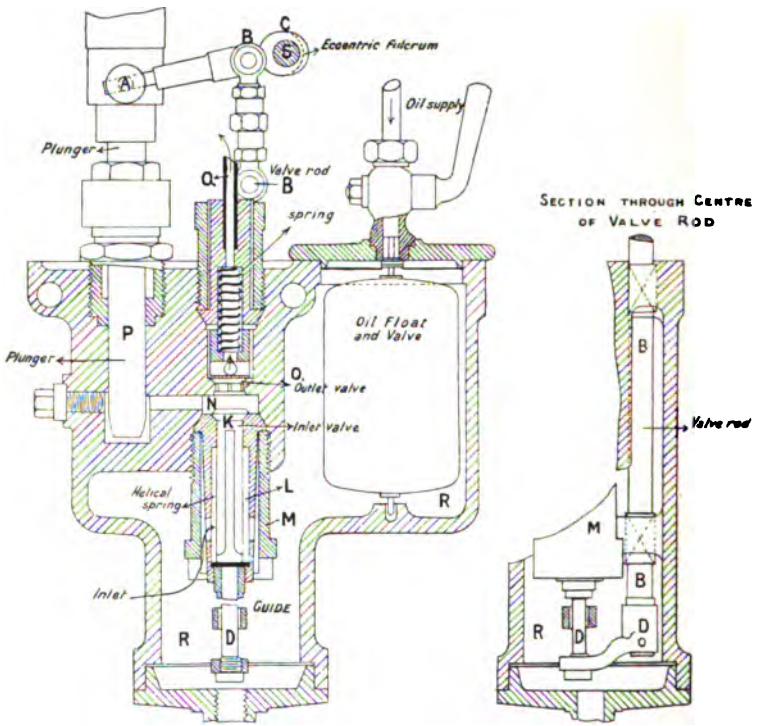


FIG. 400.—Diesel Oil Fuel Pump.

the heat required to convert the liquid into vapour is supplied by the working substance and, consequently, tends to reduce the temperature of combustion. The comparatively slow combustion is principally due to this cause.

The oil fuel is injected into the cylinder by a pump, Fig. 400, the plunger *P* of which has a constant stroke and is driven from the end of the valve cam shaft (*T* in Fig. 393). *ABC* is a rocking lever with a fulcrum *C*, and is actuated by the movement of the

plunger *P*. The valve rod *BB* is worked by the rocking lever, and, by means of the arm and spindle *D*, opens the oil inlet valve *K* against the action of the spring *L*. The fulcrum *C* is fixed eccentrically on a shaft *S* which is caused to rotate, but not to make a complete revolution, backwards and forwards as necessary, by the action of the governor. The action of the governor alters the stroke of the rod *BB* and thus controls the quantity of oil passing through the inlet valve *K* into the small chamber *N*, whence it is forced by the pump through the outlet valve *O* to the pipe *Q* and to the oil fuel valve (*E* in Fig. 392) in the head of the cylinder. The oil float automatically regulates the level of oil in the float chamber *RR*.

In Fig. 401 four indicator diagrams are shown, taken under

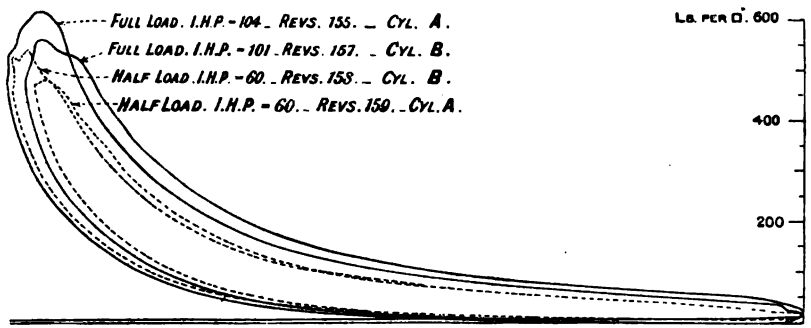

FULL LOAD. I.H.P. = 104. REVS. 155. CYL. A.
FULL LOAD. I.H.P. = 101. REVS. 157. CYL. B.
HALF LOAD. I.H.P. = 60. REVS. 158. CYL. B.
HALF LOAD. I.H.P. = 60. REVS. 159. CYL. A.
LB. PER □". 600
400
200

FIG. 401.—Indicator Diagram of Diesel Engine.

various conditions from the two cylinders *A* and *B* of a Diesel engine of 160 Brake H.P. The principal difference between these and the diagram shown in Fig. 377 is that in the Diesel engine, although the compression is higher, there is a continued high pressure for a greater proportion of the working stroke.

The following test of a 500-B.H.P. 3-cylinder engine, working at 150 revolutions per minute, is taken from a small pamphlet issued by the Diesel Company :—

	Full Load.	Half-Load.
Oil consumed per I.H.P.-hour	0·3202 lb.	0·2828 lb.
,, ,, B.H.P.-hour	0·4103 lb.	0·4196 lb.
Mechanical efficiency	72·3 per cent	...
Thermal efficiency	39·6 per cent	44·9 per cent

In the above table the mechanical efficiency is low compared with a steam engine, but it must be taken into account that in a Diesel or other oil engine the boiler efficiency is unity and the comparison is really in favour of an internal combustion engine; thus—

$$\text{Overall efficiency} = \frac{\text{work done per minute}}{\text{fuel energy expended per minute}}.$$

With the most economical form of steam engine, at least 1 lb. of the same kind of fuel must be burnt to obtain 1 I.H.P., and, as will be seen from the above table, less than one-third this weight is required for a Diesel engine.

The governor is of the usual centrifugal type, and is worked by gearing from the crank shaft as shown in Fig. 392. A full supply of air enters the cylinder at every stroke and is not affected by the action of the governor.

Mirlees-Diesel Marine Engine.—The general arrangement of a 120-B.H.P. oil engine running at 400 revolutions per minute is shown in Fig. 402. The engine runs in one direction only, and reversal is obtained by transmission gearing. Several engines of this type have been supplied to the Admiralty for ship-lighting and boat propulsion.

Diesel engines, which are reversible, have recently been constructed to work on the two-stroke cycle. It is claimed for this type that (*a*) the efficiency is equal to that of the four-stroke cycle, (*b*) a saving of over 25 per cent in weight is easily obtainable, and (*c*) reversal is obtained in a very simple way without serious loss of mechanical efficiency when applied to marine propulsion.

Vosper & Co., of Portsmouth, have supplied an oil-driven launch to the Admiralty, which is stated to have satisfied all requirements of speed, power, and manœuvring qualities.

Advantage of High Compression.—The range of temperature in the cylinder of a gas or oil engine is evidently between the highest temperature obtained by the combustion and that of the exhaust gases. Considered as a perfect heat engine, the

$$\text{Efficiency} = \frac{T_0 - t_0}{T_0} = 1 - \frac{t_0}{T_0}, \text{ using the usual notation.}$$

The lower temperature t_0 is practically governed by that of the

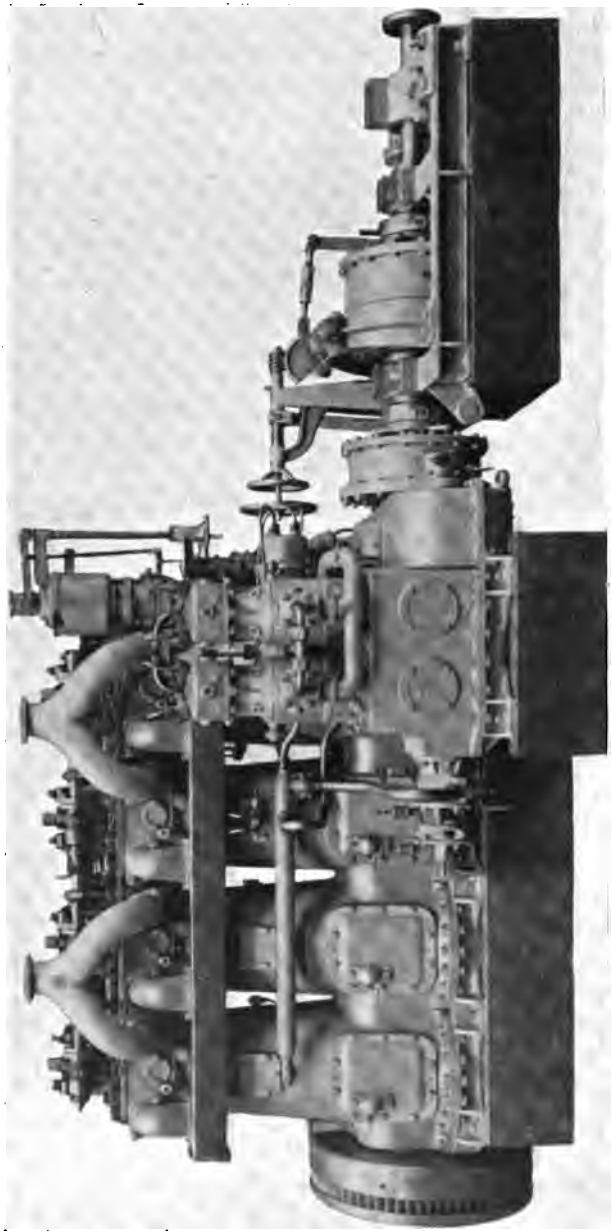


FIG. 402.—Mirlees-Diesel Engine for Boat Propulsion.

atmosphere, and an increased efficiency is only obtainable by increasing T_0, the initial absolute temperature of the working substance. It is generally considered that T_0 varies directly as the compression; but this is incorrect, although the tendency is to move in that direction. In practice the greater efficiency is more correctly attributable to the better maintenance of pressure after the beginning of the stroke, and not to a peak of pressure obtained by a peak of high temperature. High compression is usually obtained by a small clearance in the cylinder behind the piston and accurate fitting of the piston and valves of the engine.

The cylinder and working parts of an engine must be constructed for the maximum stress obtained, whether this stress is attributable merely to a peak of pressure or to a pressure maintained for a fairly high fraction of the working stroke (as in the Diesel engine).

The only objection to high compression is when working with low flash-point gases and oils. There is then great liability to pre-ignition and back-firing with reversal of direction of motion of the shaft, and a commensurate increase in the working stress imposed on the engine and its connections.

Electrical Transmission.—This may be adopted for marine purposes to attain either one of two distinct ends: that is, either as a means of entire transmission from one power to another, or simply as a means of obtaining reversal. In the former case, it has been proposed to run steam turbines at very high speeds, which ensure the greatest economy of fuel, to drive electric generators, which in turn will drive electric motors connected by shafting with the propeller. By this method, dynamos running at many hundreds of revolutions per minute will by means of electric motors drive screw-propelling shafts at, say, less than one-tenth the rate of revolution of the dynamo. Weight will be saved by adopting the higher speeded turbine, but some will be added by the generator and motor. Higher efficiency and economy of fuel will be obtained by the turbine, but some will be lost in transmission both in the generator and motor. In the Parsons type of steam turbine applied directly to marine propulsion, the resistance opposed by the thrust-block friction is almost eliminated; this resistance in ordinary reciprocating engine practice is by no means a negligible quantity, but with entire electrical transmission, and the possibility of obtaining increased propeller efficiency of a low-speeded propeller over one driven at high speed directly by a turbine, the combined

higher efficiency may possibly balance the thrust loss. There is sufficient material knowledge in existence, both from actual trial and from experiment, to be able to form a fairly reliable estimate of the claims set forth by the exponents of entire electrical transmission.

Entire electrical transmission is becoming more increasingly adopted for petrol cars and heavy lorries on the Continent, and is also in use in connection with the "Diesel" oil engine, and, it is believed, with some measure of success; but a *partial* system, invented by M. C. del Proposto, appears more promising.

The *del Proposto system of reversing* does not necessitate the full power of the engine being absorbed by the main generator (or dynamo), but only such proportion of it as may be required for

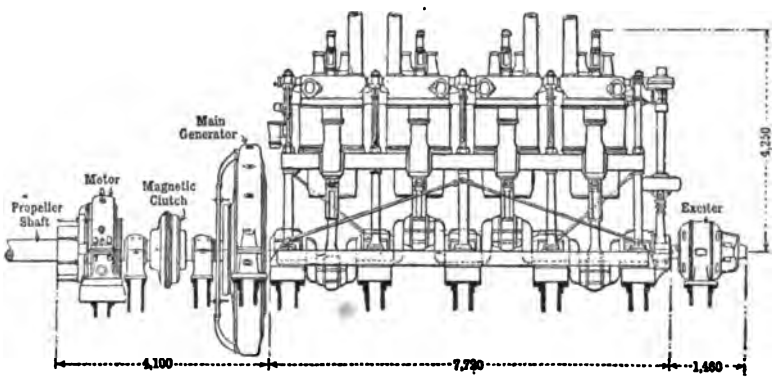


FIG. 403.—Del Proposto-Diesel Electrical Transmission.

reversing. For steam turbine engines, about one-third *power*, giving about 70 per cent of full *speed* astern, is generally found sufficient, and presumably the same would apply to Diesel or other engines. In the del Proposto system, Fig. 403, there is an exciter dynamo driven by an extension of the engine shaft at the forward end. At the after end is the main generator. In connection with the shaft coupling, and between this generator and the electric motor for reversing, is the magnetic clutch. At full ahead or normal speed the clutch is engaged by using the current supplied by the exciter dynamo; and, the brushes being raised, the main generator and reversing motor are not excited. It is stated that the energy required for working the exciter dynamo is very small compared with the whole output of the engine, and is almost

negligible. In an outline sketch of this arrangement, the main dimensions are about 43½ feet long and 21 feet high for 1000 horse-power. The main electric generator, the armature of which acts as a flywheel, is about 13½ feet overall diameter, and thus brings the level of the shafting rather high in the engine-room. Improvements are being made and may be obtained with this method of reversing; but for the present, attention can only be drawn to the essential difference between entire electrical transmission and the del Proposto system of a magnetic clutch engaged by means of an exciter dynamo for full power ahead driving, and with a moderately powered dynamo for reversal.

Reversing.—Any engine used for marine propelling purposes must be capable of reversal of direction of motion, taking an Admiralty standard, in less than 20 seconds when working at full speed and full power. In actual practice from 7 to 10 seconds generally suffices for steam engines. The overall efficiency is reduced by 10 to 60 per cent by using transmission gear, and is accompanied by an undesirable increase of complexity and liability to break down. Such expedients as a feathering propeller or a reversing clutch are not applicable to large-powered engines with either safety or efficiency, although useful for small-power installations. Two methods remain, both capable of development on practical lines: (*a*) electrical transmission, as described above or otherwise; and (*b*) direct reversal of the engine shaft as in ordinary steam piston engines, with no constant loss of efficiency.

The following systems of reversing may be mentioned as having been tried with more or less success in small-powered engines, and which are, perhaps, best considered as still experimental:—

1. By moving the timing arrangement to alter the instant of ignition, so making the explosion very much earlier in the compression stroke, and by thus checking the piston, the engine backfires and reversal of the engine shaft is obtained after a few strokes of the piston. This system tends to strain the engine, but has been in successful use for two or three years on some motor cars, in which the engines use paraffin and work on the two-stroke cycle. It appears to be applicable to large-powered engines using electrical ignition.

2. Two heart-shaped cams of the slot type, combined with the timing arrangement and a tumbler running in the slot, are used in the Ericson motor. This system appears unsuitable to any but the

smallest engines owing to the frictional and mechanical difficulties, but it is capable of further development.

3. A system of clutches fitted on the cam shaft allowing the valves to be operated by either *ahead* or *astern* cams fitted on the same cam shaft. In this the number of working parts is large, but they are of a simple character.

4. By altering the lead of the cam shaft relatively to the engine shaft, and thus bringing the cams into a position suitable for the reverse direction of the engine shaft. This has been adopted in the Thornycroft paraffin engines supplied for some Italian submarine boats, the starting of which is effected by compressed air.

5. In the case of the Diesel engine where compressed air is stored, and used when necessary for starting the engine, it is possible to use this compressed air to check the motion and bring about the reversal of the engine shaft when the fuel is cut off from the cylinder. Having obtained reversal in this way, the fuel can be again brought into play and motion continued by combustion as before, it being a matter of indifference in which direction the crank revolves so long as the piston and cylinder do not change their relative motion. Only a very slight alteration in the lead of the cams is necessary to effect this. The storage reservoirs are an additional feature and would necessarily be of large capacity, but this in itself presents no great mechanical difficulty of arrangement.

Relative Advantages and Disadvantages of Oil Engines.—1. Compared with the best steam engines using either coal or oil, the fuel consumption is about one-half; but the economy decreases with the speed and power in rather greater ratio than that of steam engines, because the jacket loss is nearly constant at all powers. A great increase of power distance is possible even in comparison with the best type of Parsons steam turbines.

2. No great lengths of steam pipe to warm through, with consequent loss from radiation and trouble from leakage.

3. Less delay in starting, compared with steam-raising, in lighting fires, and warming engines.

4. Less labour and cost of upkeep, but only with increased intelligence of a trained staff.

5. For generating electrical energy for transmission to, and for working, guns and torpedoes, and for lighting purposes, the time required to get ready is reduced, and by concentration of the motive power into comparatively large units the economy is increased.

6. If the oil used is of a safe flash-point there is no difficulty in easy and safe stowage, and the capacity may be increased without increase of the space required for coal.

7. The weight is being rapidly reduced per I H.P., and the space is already less than that required for a steam engine and boiler plant, but the concentration of power into very large units has not yet been satisfactorily obtained for marine working.

8. Reversing and starting probably present the greatest obstacle to their adoption, but do not appear insurmountable although possibly in the infancy of their development.

9. The fuel, being liquid, can be stored in compartments of any shape and at any distance from the engine. It can be pumped into or out of the ship with comparative ease and cleanliness.

QUESTIONS FROM EXAMINATION PAPERS

1. What is the generally accepted explanation of the nature of heat and temperature?

Define the "Specific Heat" of a substance. If the specific heat of air be ·2375, how many cubic feet of air can be raised one degree in temperature by one British thermal unit, the specific volume of air being 12·38 cubic feet at 32° F.? *Ans.* 52·13 cubic ft.

2. Mention the ways in which heat is transferred from one point to another in a body, and illustrate your answer by reference to the transference of heat from the furnace of a boiler to the steam formed in it.

3. Define the term "Total Heat of Formation" of steam at one temperature from water at another temperature.

Obtain numerical results for the total heat of formation of steam from water at 100° F., at

(1) 380° F. (1) *Ans.* 1128 b.t.u.
(2) 160° F. (2) *Ans.* 1062 b.t.u.

What is meant by the dryness fraction of steam, and how would a dryness fraction of 0·9 affect your answer to (1) above? (3) *Ans.* 1043 b.t.u.

4. For any type of Vertical Inverted Engine with which you are acquainted, sketch and describe two of the following detail parts :—

(*a*) A large piston rod, stuffing-box with metallic packing;
(*b*) The piston-rod slipper and guide;
(*c*) A piston slide-valve;

making special reference to the provision made to allow for the wear of parts.

5. Define the term "the Latent Heat of Steam."

20,000 lb. of steam per hour are discharged from an engine to a surface condenser and there condensed to water at a temperature of 100° F. Find the number of lb. of circulating water initially at 60° F. that must pass through the condenser per hour in order that the temperature of the condensing water on leaving the condenser may be 85° F., the latent heat of steam at 100° F. being 1045 b.t. units.

6. State the three ways by which heat may be transmitted from one body to another, explaining clearly how the methods differ.

In a boiler, the heat from burning coal eventually appears as heat in the steam leaving the boiler. By which of the above methods has the heat been transmitted from the coal to the steam?

7. By means of line sketches indicate how a large ship is steered by working

a steering wheel on the upper deck, and sketch and describe in detail the "differential gear" which causes the motions of the steering engine to follow those of the steering wheel.

8. Sketch and describe any type of water-tube boiler suitable for a battle-ship, naming the type you choose.

9. For what reason is a vacuum maintained in the condenser of a steam engine? Sketch and describe the pump by which this vacuum is produced, stating the various methods by which these pumps are driven in naval vessels.

10. A war-vessel has a bunker capacity of 1080 tons of coal. When steaming at 15 knots she burns 170 tons per day. Find approximately the quantity of coal burnt per day at 10 knots; and if the full speed of the vessel be 19 knots, find the number of nautical miles she can steam at full speed, starting with bunkers full.

11. By means of sketches show the positions of the common D slide valve relatively to the cylinder ports, and the position of the piston in the cylinder (*a*) when admitting steam, (*b*) when cutting off steam, (*c*) when the piston is at one end of its stroke. Mark the directions of motion of both the valve and piston on each sketch.

12. Define the terms "pitch ratio," "pitch angle," "projected area," and "slip," as applied to screw propellers.

Obtain a formula expressing the percentage of slip in terms of the pitch, revolutions of the propeller, and actual speed of ship.

13. Sketch in plan the engine room of a large ship, showing in outline the main and all the principal auxiliary engines that would be situated in it. Name each engine and state briefly its use.

14. Give an account of the addition of heat to water, and formation of steam under constant pressure. Define the terms—

Volume per lb. of steam;

Relative volume of steam;

and quote the results for atmospheric pressure.

15. Describe the construction of an ordinary surface condenser, giving details of the jointing of the tubes in the tube plates, and the tube plates to the shell.

16. Sketch and describe a single-acting air pump as usually fitted in connection with the main engines of a ship, showing clearly the various sets of valves and the means of working the pump from the main engines. What advantages has a separately driven pump for this purpose?

17. Define a "British thermal unit," and state what is meant by its "mechanical equivalent."

An engine uses 20 lb. of water per I.H.P. per hour, the feed temperature being maintained at 100° F., and the boiler temperature at 360° F. What percentage of the heat taken up in the boiler is converted into work, the total heat of steam at 360° F. being 1190?

18. Solid bodies expand generally when their temperature is raised; describe an experiment to illustrate this phenomenon, and mention two instances in which provision is made to allow for this in the propelling machinery installation of a ship, giving details.

19. Define work, and power, and investigate the formula for the indicated horse-power of an engine—

$$\text{I.H.P.} = 2\,\frac{\text{PLAN}}{33{,}000}.$$

A steam cutter has a two-cylinder simple engine. Diameter of cylinder, 6 inches; stroke, 6 inches. If the mean effective pressure be 50 lb. per square inch, and the revolutions be 400 per minute, estimate the I.H.P. developed.

20. Sketch two of the following details:—

(*a*) A section through an air pump;
(*b*) A section through a boiler stop valve;
(*c*) A crosshead bearing;

giving a brief description of the parts as illustrated by your sketches.

21. Describe, with sketches, the detail of fitting of the screw propeller to the stern shaft, showing in *outside* view only the relative position of the stern bracket bearing.

22. It is generally assumed that the coal consumption for a ship is proportional to the I.H.P. developed by the engines; show that then the coal consumption for a certain voyage will vary as the square of the speed at which it is made.

23. State how you would take an indicator diagram and work out the mean pressure from it.

What are the cylinder constants?

24. Describe in general terms the feed-water arrangements for the boilers of any ship you know, stating how losses are made good and storage provided for, and detailing the pumps, suctions, and deliveries in connection with the system.

25. Describe, with sketches, the furnace fittings of any boiler with which you are acquainted; and explain how the rate of combustion of fuel on the fire-grate can be varied.

26. By means of an outline sketch show the arrangement of parts of a return tube cylindrical boiler. Letter the various parts, and give the corresponding names.

27. Describe, with detail sketches, any two of the following:—

(*a*) Method of fitting plain and stay tubes in a cylindrical boiler;
(*b*) Method of staying the front to back plate in the steam space and in the space between the tubes and furnace;
(*c*) Method of staying the top plate of a combustion chamber;
(*d*) Method of attaching the furnace to the combustion chamber and boiler front plate.

28. Why is it necessary to have a guide for a piston-rod crosshead? Show how a separate guide is required for ahead and astern motion.

29. Describe and sketch one of the following:—

(*a*) The method employed to enable a piston to work steam-tight in a cylinder;
(*b*) The stuffing box and packing for a large piston rod.

30. If the conditions of working a turbine were initially saturated steam, temperature 380° F., and finally steam of temperature 120° F. and dryness 0·75, how much heat is absorbed by the turbine per lb. of steam?

31. Sketch three of the following details of boilers:—

(1) The connections of the elements of the Belleville boiler with the feed collector and steam drum;
(2) The supporting frames and casings of the Babcock and Wilcox boiler;
(3) Furnace of a White-Forster boiler arranged to burn oil fuel;

(4) The tube end connections to headers of the Niclausse and Dürr boilers.

32. Describe Howden's system of forced draught, and in *outline* only sketch the arrangement with any boiler with which you are acquainted.

33. Show, by sketches, how you would arrange the furnaces and combustion chambers of a double-ended boiler, and explain how the different parts of the combustion chambers are stayed.

34. With the aid of a sketch, show what is the usual position of the working water-level in a cylindrical boiler, and explain the probable effect if the water-level is (*a*) much too high, (*b*) much too low.

35. What is meant by the statement that "the vacuum is 25 inches, while the barometer shows 30 inches"?

Assuming the pressure of the atmosphere to be 15 lb. per square inch, what is the pressure corresponding to 25 inches of vacuum? *Ans.* $2\frac{3}{4}$ lb. abs.

36. How are the tubes in a cylindrical boiler fitted? What means are taken to protect the combustion chamber ends?

37. What is meant by the *heating surface* of a boiler? Name the parts of a cylindrical boiler which form the heating surface.

38. Describe, with the aid of a sketch, the construction of a Belleville boiler, indicating the course which the feed water has to take.

39. Describe the path of the water from the feed pump of a Belleville boiler (economiser type) to its exit as steam into the steam pipe; also the uses of the various parts through which the water passes *during* its conversion into steam.

40. What are the distinguishing features of water-tube boilers as a class? How, and for what objects, were "water walls" formed in many boilers of the curved-tube type, such as Thornycroft's.

41. Name the various kinds of water-tube boiler of which you have any knowledge, dividing them into *curved*-tube and *straight*-tube boilers, and describe any example of the former class.

42. Explain how a boiler is liable to suffer from undue haste in raising steam, and describe the precautions that are necessary when steam is being raised. Mention the various mountings on a boiler, and their uses.

43. Describe the construction and use of the hydrometer, and state how the service hydrometer is graduated.

If two pints of sea-water are boiled until one pint remains, what is the density of the remaining water on the service hydrometer? *Ans.* 20°.

44. Describe the construction and action of the ordinary steam-pressure gauge.

45. Describe the closed stokehold system of forced draught.

How is the air pressure measured?

46. An engine overcomes a resistance of 44,000 lb. through a distance of 1500 feet in $1\frac{1}{2}$ minutes: what horse-power is represented by this performance of work? *Ans.* $1333\frac{1}{3}$ I.H.P.

47. What is the object in cutting off the supply of steam to a cylinder before the end of the stroke? Calculate the mean forward effective pressure in a cylinder using steam of 200 lb. pressure per square inch by gauge, if the cut-off is at one-third stroke and the back pressure by gauge is 90 lb. per square inch. *Ans.* 50 lb.

48. Compare the advantages of a triple expansion engine with a three-

cylinder engine of the same power, each of whose cylinders takes steam directly from the boilers.

49. Steam of 160 lb. absolute pressure is admitted to a cylinder and cut off at quarter-stroke: find the average effective pressure for the stroke, the back pressure being 10 lb. absolute. *Ans.* 88 lb. per square inch.

By reference to the above example, show the advantage gained by using steam expansively.

50. Steam entering a cylinder is cut off at three-quarter stroke, the final pressure is 150 lb. per square inch absolute: what is the initial pressure (*a*) by gauge, (*b*) absolute? *Ans.* 185 lb., and 200 lb.

51. What are the two important properties of steam which are made use of in connection with a set of marine engines? Explain how they add to the efficiency of the engines.

52. Describe and give an outline sketch of the "all round" variety of link motion.

53. What are the advantages of corrugated furnaces? Describe how you would fit this form of furnace to the combustion chamber.

54. Describe how an abestos-packed cock is repacked. Sketch one so fitted.

55. Sketch a safety valve for a high-pressure boiler, showing in detail the lifting gear. What amount of lift should be given to the valve?

56. Describe any method of inducing a draught up the funnel. What are the advantages of induced draught over the closed stokehold system?

57. Estimate the temperature of the gases passing from the furnace when coal capable of yielding 14,000 b.t. units per pound is burnt with 19 lb. of air, supplied at a temperature of 75° Fahr., supposing half the heat of combustion to be radiated to the furnace crown and sides.

58. Explain fully the advantages of fitting vessels with closed ash-pits, and state the parts of the machinery and boilers which you consider would require special attention when forced draught is being used.

59. What increase of power would you expect to obtain from a set of boilers fitted with forced draught, compared with the same set under natural draught?

60. A set of six boilers placed back to back are required for a ship. Show by an outline sketch the general arrangement of boilers, smoke boxes, uptakes, and funnels.

61. Flame is sometimes seen near the mouth of the funnel: what causes this appearance? Is it beneficial or detrimental? Why so?

62. Distinguish between saturated and superheated steam. What percentage of water is carried in suspension by steam which has had 1100 b.t. units expended per pound in its production at a temperature of 347° Fahr. from water at 75° Fahr.?

63. When steam expands in a cylinder, describe how the pressure varies with the increase of volume. Point out the distinction between this, and the expansion of a gas by Boyle's Law, and describe the variation of temperature which a steam cylinder undergoes in the double stroke.

64. State an expression for the efficiency of a thermally perfect engine. If an engine working between the limits of temperature 400° and 100° requires an expenditure of 200 b.t. units per minute per horse-power, determine the relative efficiency.

65. What is meant by the "total heat of formation of steam"? Feed water

at 62° Fahr. is pumped into a boiler and evaporated into steam containing 2 per cent of moisture at a pressure of 300 lb. (422° Fahr.). What quantity of heat is added to each 1 lb. of water in the process? What additional heat would be required to superheat the steam to 500° Fahr.?

66. Give the formula for calculating the calorific value of a fuel, and explain how the formula is obtained. What is meant by the theoretical evaporation of a fuel in pounds of water, from and at 212° Fahr.?

67. In a boiler 22 lb. of air are supplied for 1 lb. of fuel, and the funnel gases escape at a temperature of 800° Fahr.: find the heat wasted per 1 lb. of fuel when the temperature of the air supplied is 65° Fahr., and the specific heat of the funnel gases is 0·23°. Find the maximum efficiency under these conditions.

68. A fuel containing 87 per cent of carbon and 13 per cent of hydrogen is completely burnt in the furnace of a boiler; the analysis of the funnel gases by weight, neglecting any steam passing away with the gases, shows 13 per cent of CO_2, 9·4 per cent of O_2, and 77·6 per cent of N_2. Calculate the composition of the funnel gases when taking the water vapour into account, and find the number of pounds of air supplied per 1 lb. of fuel.

69. Describe either the Klinger form of gauge glass, or a plate water gauge.

70. Define "British thermal unit" and "foot-pound," and state the relation between these units. Calculate the heat, in b.t. units, equivalent to the work done in three hours by an engine developing 15 horse-power.

71. What is the boiling-point of water at atmospheric pressure, and how does the boiling-point vary with a rise of pressure? What is the amount of latent heat, sensible heat, and total heat of formation of 1 lb. of steam at 185° Fahr., from water at 38° Fahr.?

72. Make a longitudinal sectional elevation of a cylindrical boiler, showing the general arrangement of the various parts, and, as fully as you can, the various forms of joint adopted for the internal parts.

73. Is the steam jacket as fitted to marine cylinders of more use with the ordinary single-cylinder engine or a stage-expansion engine? Why?

74. Explain the terms "maximum density of steam" and "saturated steam." Calculate the "dryness fraction" of steam formed from water at 90° Fahr., at a constant absolute pressure of 225 lb. per square inch (temperature 392° Fahr.), by the expenditure of 1100 b.t. units per 1 lb.

75. What data are necessary in calculating the horse-power of a marine engine? What is the value in "inch-tons per second" of one horse-power? How is the horse-power of a marine steam turbine obtained?

76. What are the chief constituents of coal which give it its value as a heat-producer? Describe the chemical actions which take place during the combustion of coal, and state an expression for the heat value of coal.

77. Explain the formation of smoke and flame.

78. Explain the action of a chimney, or funnel, in inducing a draught. State the amount of air required per 1 lb. of coal burnt, and mention the losses which occur—first, when the supply of air is limited; second, when the air supply is too great.

79. What are the advantages of using superheated steam, and where is the best place in which to apply the superheat?

80. Sketch an athwartship section through a main engine bedplate of a large ship, and show how the bedplate is secured to the engine bearers.

81. The cylinders of a compound engine are 12 and 24 inches diameter, and the mean effective pressures are 70 and 20 lb. per square inch respectively; the stroke is 9 inches, and the revolutions are 100 per minute. Find the horse-power. *Ans.* 77 I.H.P.

82. An ash-hoist engine has two cylinders of 6 inches diameter and 6 inches stroke; the mean effective pressure in each cylinder is 60 lb., and the engine makes 2 revolutions per second: what is the horse-power? *Ans.* 12·4.

If all the horse-power indicated be utilised, at what rate will the engine raise a bucket of ashes weighing $2\frac{1}{2}$ cwts.? *Ans.* 24·8 feet per second.

83. The initial steam pressure in a cylinder is 100 lb. by gauge, and cut-off is at one-third stroke: what is the average effective pressure on the piston if the steam is exhausted to the atmosphere? *Ans.* 68 lb. per square inch.

84. What is meant by using steam expansively? Show, in any way, that expansive working should be advantageous in point of economy.

85. Give an account of the action of the cylinder metal leading to waste of heat in the steam engine, and show how the compound method of expansion leads to increased economy. What fittings in connection with the cylinder tend to reduce the above waste, and in what way?

86. What is the effect of the receiver pressures in a triple expansion engine?

What is the effect of making the cut-off in each of the cylinders earlier respectively? What are the means provided for doing this?

87. Explain how the condenser tube ends are made water-tight in the tube plate; and what is the result if they leak? Describe briefly the use of the air pump and the use of the feed tank.

88. The diameter of a cylinder is 30 inches, and the mean effective steam pressure on the piston is 80 lb. per square inch when developing 1600 I.H.P.: at what speed is the piston moving? *Ans.* 933·3 feet per minute.

89. Why is it better to carry out a large expansion of steam in stages, as in the three-stage expansion engine, rather than in one stage, as in the simple engine?

90. Name, and give the use of, the various materials used in the construction of a set of engines and boilers.

91. Describe the arrangement of stuffing box now usually fitted for piston rods. Describe the various parts of which a piston for a large engine is composed.

92. With the aid of a sketch, show how a main bearing is formed. What is the object of the construction in this and other working bearings?

93. Describe the bed or sole plate of a set of vertical engines, and how the cylinders are supported. Why is it necessary to have a guide for the piston-rod head?

94. Describe the line of shafting from the engine to the propeller, naming the various bearings and their object. Why is the shafting made hollow?

95. Describe briefly, with the aid of a simple sketch, the construction of a cylinder, indicating how the steam enters and leaves.

Name the mountings usually fitted to the cylinder.

96. What is the object in fitting a steam jacket to a cylinder? Describe how the steam used in the jacket is kept distinct from that used for driving the piston.

97. Comparing the modern with the early forms of marine engines for a given power, describe briefly the reasons for—

(*a*) The increased economy of coal;

(*b*) The saving in weight and space.

98. Describe the construction of a piston suitable for a large cylinder. How is the piston secured to the piston rod?

99. Describe the arrangement of a piston-rod stuffing box and gland suitable for a modern set of engines.

100. Sketch and describe the piston-rod crosshead bearing. Of what material are the bearing surfaces formed?

101. Describe the construction of a piston for a large cylinder, showing how it is made steam-tight in the cylinder. Name the different materials of which the various parts are made.

102. Name and state the object of the various mountings you would expect to find on a low-pressure cylinder of a compound engine.

103. Describe and explain the use of a connecting rod, giving particulars of the bearings at each end of it.

104. Name, and give the reasons for, the principal operations of the slide valve during a complete revolution of the engine.

105. Sketch a single-ported slide valve in mid position on the cylinder face, and also in the position it occupies at the commencement of the piston stroke. Mark in the sketches the *steam* and *exhaust lap* of the valve, and the *lead* of the valve. Define these quantities, and state the uses of giving lap and lead respectively.

106. Show, by means of outline sketches, the relative position of piston and slide valve at the four principal operations of the valve.

107. What are the advantages gained by the use of the piston type of slide valve? Describe the construction of one of these valves.

108. What is the object in fitting link motion to a set of engines? Explain the effect of putting the link in intermediate positions between "full gear ahead" and "stop."

109. Define the terms "lap" and "lead" as applied to a slide valve, and explain their object.

110. Describe any arrangement you know of for reversing a set of engines.

111. Explain what is meant by the "dead centre" of an engine. To what difficulty does this give rise, and how is it overcome in single and double acting engines?

112. Give sketches showing how the steam passes through the engines in three-cylinder and in four-cylinder three-stage expansion engines. Why is the four-cylinder type used in place of the three-cylinder type in some cases?

113. Give reasons for the superior economy obtained in modern engines in comparison with those formerly in use. State what coal consumption is usually obtained under good circumstances at sea (*a*) for main engines, (*b*) for all purposes at full power. Why is the consumption sometimes greater at very low powers per I.H.P.?

114. Enumerate and explain the use of the fittings attached to a large marine engine cylinder.

115. Where are reducing valves fitted on a steam cylinder, and for what reason? Describe one, and give a sketch of its action.

116. Draw the horizontal section through the exhaust port of a high-pressure cylinder of a marine engine with a cylindrical slide valve, and show how the parts are attached to the rod.

117. Show by sketches the method of jointing a liner in its cylinder, and the method of allowing for differences in expansion of cylinder and liner. Why is the method described preferable to any other well-known type?

118. Sketch a thrust block of the horse-shoe type, showing how the white metal is secured to the shoe, and how the oil supply is arranged. How is the shoe kept cool when under way? Why is cooling necessary?

119. Describe, with sketches, how consecutive lengths of propeller shafting are connected together.

120. Why is a thrust block fitted to a marine engine? Give a short description, aided by sketches, of (*a*) thrust block, and (*b*) thrust shaft. How is the thrust taken in a marine steam turbine?

121. Sketch a section through the bush fitted at the inboard end of a ship's stern tube. Of what material is the bearing surface made, and how is it secured in the bush?

122. Sketch and describe the stern-tube bearing of a large vessel. How is the stern shaft removed? Describe the necessary fittings.

123. Sketch and describe the ordinary form of piston rod and crosshead, showing how the rod is attached to the piston, and the provision made for lining up the shoe. What is the use of a stretching length, and where are they fitted?

124. Describe, with sketches, a piston and packing used for a high-pressure cylinder, and the metallic packing used for the piston rod.

125. Sketch the usual arrangement adopted for independently linking up engines, showing how the gear is made practically inoperative when going astern.

126. Show by sketches the positions of a simple slide valve on the valve face at the points of admission, cut-off, release, and compression. Indicate the direction of rotation obtained by the motion of the piston.

127. Define "open" and "crossed" rods as applied to link motion. What is the effect in each case on the "lead" when "linked up"? How is the "angular" advance obtained, and why is it used?

128. Describe the principal differences between a jet condenser and a surface condenser. Which type is now used, and why?

129. Describe a form of air pump worked by an independent engine. Sketch and describe an air pump with only one set of valves. Sketch a Kinghorn metallic valve and seating, and describe its action.

130. What is meant by the "closed exhaust" system? Explain any reason for its retention in modern vessels.

131. Describe the arrangements made for draining the steam and exhaust systems. Sketch and describe an expansion steam drain trap, and state your reasons for its action.

132. Describe the fittings and pipes attached to a feed tank. Give an outline sketch.

133. Sketch, in plan, an outline arrangement of main steam pipes of any modern twin-screw ship with which you are acquainted, indicating the position of all valves and expansion joints.

134. Sketch a flange joint of a steel main steam pipe, showing how the flange is secured to the pipe. Describe some methods of making the joint between the flanges, and state your opinion of the advantages of any particular type of joint.

135. Sketch a modern type of evaporator suitable for working with exhaust or low-pressure steam, showing as many of the mountings as you can.

136. Describe a method of keeping a constant density in an evaporator. What precautions are necessary to keep the apparatus in working order?

137. Describe the method of supporting and staying the cylinders of a large marine engine. How is expansion allowed for?

138. Name the parts passed through by the condensed steam from the air pump to the feed tank, and state the means adopted for minimising the water loss.

139. Describe the working of an evaporator, and state the various pressures in its parts when utilising steam of low pressure. What is the advantage of generating steam at low absolute pressures in evaporators, and what precautions should be taken to prevent priming?

140. Name some of the various filtering materials which are used for the purification of boiler feed-water on board ship. Why is a filter necessary? Sketch and describe any form of filter with which you are acquainted.

141. Sketch a section through the barrel of a feed pump, showing the suction valves and delivery valves, and the method of packing the water piston.

142. Sketch and describe a feed-water heater. What are the advantages of feed heating?

143. A vessel can steam 12 knots on a coal consumption of 45 tons per day; the bunker capacity is 375 tons. What would be the greatest speed at which she could steam between two ports 1536 sea-miles apart? *Ans.* 15 knots.

144. What are the principal resistances offered to the passage of a vessel through the water? Show how the relation "I.H.P. varies as the speed cubed" for moderate speeds is obtained.

145. What is the object in making provision for the independent adjustment of the cut-off in a particular cylinder of a set of stage-expansion engines?

146. With the aid of a diagram, indicate what is meant by the "angular advance" of the eccentric, and explain why this is given.

147. What is the object in fitting a double-ported valve in preference to a single-ported slide valve?

What arrangements are made for reducing the friction between the valve and cylinder face as much as possible?

148. Why are piston slide valves fitted to some cylinders and flat slide valves fitted to others?

Describe the relief arrangements fitted to a large flat slide valve.

149. How are the tubes of a condenser secured in place? Explain the object aimed at in the method you describe.

150. Explain the object and method of action of an air pump for a surface condenser, and state how the pump is worked.

151. Sketch and describe an air pump with only one set of valves. What advantage is obtained by this arrangement? Of what materials is the pump made?

152. Describe a circulating pump and its action. Make an outline sketch of the course of the cooling water through a surface condenser, and describe how you would pump out the engine-room bilge with a circulating pump.

153. About how much cooling water is required per I.H.P. for an ordinary three-stage expansion engine? And why?

154. Describe the general arrangement of condensers, air pumps, and circulating pumps as fitted in a large modern cruiser or battleship.

155. Name the fittings generally to be found on a condenser, and state their uses.

156. Give the names and describe the object of the various valves through which the steam has to pass on its way from the boilers to the engines. Why is it necessary to have a "manœuvring" valve as well as the regulating valve fitted at the engines?

157. What is the use of an "emergency valve"? Describe one, and state from where it can be worked.

158. Describe the use and position of a separator.

159. By means of a sketch, show the arrangement of main steam pipes and valves in the engine room of a battleship, giving the object and name of the different valves and fittings.

160. Describe the construction and action of a steam regulating valve, and state the arrangements made for effecting small alterations in the speed of the engine.

161. Describe, with the aid of a simple sketch, the arrangement of feed pipes and pumps you would expect to find in a large vessel.

162. Describe the fittings to be found in connection with a feed tank. Why is a hot-well tank fitted in some vessels as well as a feed tank?

163. Sketch and describe a feed-water filter or grease extractor. Where is it fitted?

164. Why is it necessary to use fresh water only with modern boilers? What means are adopted to reduce the waste of fresh water in the machinery department of a vessel to a minimum?

165. Describe the method of "making up" the feed-water loss.

166. How is the condition of the feed water for boilers, as regards freshness, ascertained on board ship? To what causes would you ascribe a gradual rise in density?

167. Describe, with the aid of a sketch, the course of the feed water from the condenser to the feed pump.

168. Describe an evaporator, and state its use. What is meant by the terms (*a*) generating steam, (*b*) gained steam, (*c*) coil drain, (*d*) vapour valve, as applied to evaporators and fittings?

169. What is a distilling condenser used for? Describe the method of working one, and the manner in which cold water can be produced.

170. If 900 I.H.P. is sufficient to give a Fleet Scout a speed of 12 knots, what power is required for a speed of 15 knots? *Ans.* 1758 I.H.P.

171. Two screws, each of 10·5 feet pitch, drive a vessel at $21\frac{3}{4}$ knots, when the engines make 250 revolutions per minute: what is the slip per cent of each propeller? *Ans.* 16.

172. Describe briefly the apparatus used for maintaining a supply of fresh water for boilers and drinking purposes on board ship.

173. Name and briefly state the object of the various auxiliary engines usually fitted in a battleship. Which of them are usually fitted in the boiler rooms?

174. Enumerate all the pumps you can think of which are fitted to a battleship, and state their uses.

175. Describe the course of the steam after leaving the boiler stop valve to its return as water to the feed tanks in a four-cylinder three-stage expansion engine.

176. Describe the features of the modern screw propeller. Of what material is it made, and why?

177. Describe the construction and object of an eccentric. Show by means of a simple diagram the position of the eccentrics on the shafting to produce ahead or astern motion.

178. If a ship burns 200 tons in steaming 800 miles at 11 knots' speed, how much will she require for a distance of 1200 miles at 10 knots, excluding coal required for auxiliary purposes in each case? *Ans.* 248.

179. Describe fully a method of determining the pitch of a screw propeller.

180. In a certain vessel a speed of 30·4 knots was obtained with 387·0 revolutions per minute; the pitch of the propellers is 9 feet 4 inches. What was the slip per cent? What revolutions would be required to give a speed of 32 knots? *Ans.* 15 per cent; 408 revolutions.

181. Describe briefly what is meant by "cavitation." How has its effect been reduced? And what causes tend to limit the increase in blade area of naval screw propellers?

182. A ship, developing 3000 I.H.P., steams 12 knots on a coal consumption of 2·24 lb. per I.H.P. per hour. What is the shortest time in which she could make a port 900 miles distant with 400 tons of coal available? What would be the corresponding approximate coal consumption per I.H.P. per hour, and the approximate I.H.P. developed?

183. Explain clearly what is meant by the "clearance volume" of a cylinder.

184. How is the total ratio of expansion affected by clearance? Define the "ratio of expansion" as applied to a state-expansion engine.

185. Construct as accurately as you can the elementary indicator diagram of an engine working under the following conditions:—

(*a*) Steam pressure at the engines, 129 lb. per square inch by gauge;

(*b*) Cut-off at one-third stroke;

(*c*) Absolute back pressure (mean), 10 lb. per square inch.

186. Describe an indicator, showing how the vertical travel of the pencil is assured. What precautions are required to obtain a correct indication of the pressure in cylinders?

187. Describe, with sketches, the hunting gear of a steering engine. Describe the gearing connecting the steering engine with the rudder head. How is screw gearing made self-holding, and why is it necessary to allow a certain amount of fore-and-aft play of the shaft in its bearings?

188. Describe Brown's telemotor controlling gearing. Sketch any form of joint which you would think suitable for the hydraulic piping.

189. Explain the principle on which ice is made by an ice machine of the CO_2 type. Sketch any system of producing ice with which you are acquainted.

190. Describe, with sketches, the best method you know for lubricating the crank bearings of a large engine. How would you lubricate the other principal bearings? Describe the syphon lubricator, and give the necessary precautions for adjusting its proper working.

191. Sketch and describe a valveless pump such as is commonly employed for forced lubrication. Describe the arrangement for lubricating each principal bearing.

192. A twin-screw ship, having two main groups of boilers, is under way with one group of boilers alight; orders are given to raise steam for full speed.

Explain how you would proceed to raise steam in, and connect up, the other group of boilers, and mention the precautions you would take.

193. Describe the nature of the examinations you would make before taking over the duties of Officer of the Watch in the engine room of a modern large vessel under steam, with water-tube boilers.

194. Describe fully the examinations you would make of a set of marine engines after a long voyage under steam.

195. What precaution is particularly necessary before opening out the steam valve of a water-tube boiler?

196. Describe a form of reversing gear, and the process of warming engines and getting under way.

197. Name the various kinds of water-tube boiler of which you have any knowledge, and divide them into classes—one with nearly vertical tubes and the other with nearly horizontal tubes. Describe any one example.

198. How is the independent expansion of the tubes allowed for in the following boilers respectively?—Belleville, Babcock, Niclausse, Dürr, Yarrow, Thornycroft, and White-Forster.

199. Describe any method of atomising liquid fuel with which you are acquainted. What advantage is gained by heating the air supplied for combustion? How is the water removed from the oil? Why is it necessary to employ brickwork?

200. Describe, with sketches, a Parsons marine turbine. Explain how the thrust of the propeller is taken, how the main bearings are fitted and lubricated, and how the shaft is packed where it leaves the turbine cylinder.

201. What arrangements are made to produce a better economy at low powers in a Parsons turbine propelled vessel? Why is the turbine generally less economical at low powers than a reciprocating engine?

202. At what speed of periphery would you expect a turbine to revolve when economy of fuel is obtained? And why?

203. Sketch and describe a method of attaching the blades in a turbine engine. Explain how you would determine the horse-power of a turbine engine when used (*a*) for propulsion of a vessel, (*b*) for propulsion of an electric generator.

204. Sketch and describe a method of producing astern motion of turbine-propelled vessels.

205. Describe a method of aligning the piston, rod, and guide.

206. How would you test the alignment of the crank-shaft and propeller-shaft bearings?

207. Describe the method of fitting and adjusting a crank-head bearing. How much clearance would you allow for a 16-inch pin (*a*) at the crown, (*b*) at the horns?

208. When the engines are suddenly eased down, how would you check the generation of steam in your ship?—naming the type of boiler.

209. Describe how you would test the feed water for acidity, and for quantity or proportion of sea-water mixture.

210. Show, by sketches, the method of plugging a damaged tube in an upright water-tube boiler.

211. Describe the system of firing when burning North Country and Welsh coals respectively. How can you prevent flaming and decrease smoke?

212. How would you find the steam consumption per hour from an

indicator diagram? From which cylinder would you expect the most accurate result?

213. What effect has "linking up" the L.P. valve gear on (*a*) the economy of steam, (*b*) the relative power developed in the several cylinders, (*c*) the total power of the engine? Why?

214. Describe how you patch a corrugated furnace. If the original thickness is $\frac{3}{4}$-inch, what should be the thickness of the patch? And why?

215. A funnel is 90 feet high above the fire-grate: about how much coal per square foot of grate can be burnt without using mechanical draught?

216. Explain a method of analysing the funnel gases and recording the proportion of CO_2 present in them. If the proportion is found to be 20 per cent, what would you do?

217. In using superheated steam, where would you look for possible damage to the superheater and to the machinery?

218. Describe the method of fitting and adjusting new restricted packing rings into a piston and cylinder, and state the clearances you would allow for bronze rings working on steel liners.

219. If a steam loop appears in the H.P. indicator diagram, what would you do? How does a steam loop affect (*a*) the practical working of the engine, (*b*) its thermal efficiency, (*c*) its mechanical efficiency?

220. Describe a method of measuring the shaft horse-power of a large marine engine, and state the necessary particulars for its determination.

221. Show, by sketches, the best method you know of cutting oil grooves in a crank bearing. What is an oil-containing strip?

222. Explain why a high vacuum produces better economy in a turbine than in a reciprocating engine. Sketch one method of increasing the vacuum in turbine practice.

223. Explain the system of ignition employed in (*a*) Hornsby oil engine, (*b*) Diesel oil engine. In a Diesel engine (four-stroke cycle) the mean pressure is 48 lb. per square inch for each of four cylinders, diameter 20 inches, length of stroke 50 inches, revolutions per minute 240; estimate the I.H.P.

224. Describe the process of opening, clearing, closing up, and testing a water-tube boiler as fitted in your ship or in a steam pinnace. State any particular precautions you would take during the process.

225. As Engineer Officer of the Watch, what would you do (*a*) if a bearing becomes heated, (*b*) boiler runs short of water, (*c*) feed pump refuses duty? If the Senior Engineer comes below and gives you an order about any of these matters, who is Officer of the Watch?

226. State the advantages and disadvantages of the water-tube boiler compared with the cylindrical type. Sketch *two* of the following boiler details:—

(*a*) The connection between the feed collector and elements of a Belleville boiler, showing also the connection of the tube to the lowest junction box;

(*b*) The connection of one tube to the header in a Niclausse *or* Dürr boiler;

(*c*) The method of securing the tubes in a Yarrow boiler; how are defective tubes plugged?

(*d*) The method of baffling the funnel gases in the Babcock and Wilcox boiler;

(*e*) The method of attachment of the furnace to the front plate and combustion chamber of a cylindrical boiler.

227. A large vessel having anchored in harbour, and the main engines being finished with, detail the matters requiring attention before reporting fires out and all connections with main engines shut off.

228. Detail what you consider the matters requiring attention in firing boilers in order to obtain the maximum power from them. What steps would you take to check the generation of large quantities of smoke from the funnel, and to stop flame burning at the top of the funnel?

229. In relation to a screw propeller define the following quantities:—

Pitch, diameter, length, angle, blade area, and projected area.

With a certain pattern of screw propeller a slip of 20 per cent had been found to give good results. What should be the pitch of such a screw to drive a ship at 30 knots, the revolutions being then 350 per minute? *Ans.* $10\frac{6}{7}$ feet.

230. Describe briefly the principle which governs the action of the Parsons steam turbine, and *generally* and *briefly* describe its construction.

231. Describe the thrust arrangement in a Parsons marine turbine, explaining the relation between propeller thrust, steam thrust on blades, and dummy piston. How is the propeller thrust transmitted to the ship?

232. Describe the transformation of the heat energy of the steam into work done on the blades in a Parsons turbine, and one method of securing the blades in rotor and stator.

233. State Charles' Law for a permanent gas, and explain the deduction of an absolute zero of temperature from this Law.

A torpedo is charged with air at a temperature 80° F. and pressure 1750 lb. per square inch; by how much will the pressure fall if the torpedo is put into the submerged tube and cooled down by the sea-water to a temperature of 50° F.?

234. Explain the difference in the compression of air when conducted isothermally and adiabatically, illustrating your reply by reference to the compression of air in a torpedo air compressor, and during the compression stroke of a Diesel engine.

235. Describe the four-stroke and the two-stroke cycles used in gas engines, and give a diagrammatic section through the cylinder of a gas engine of each type, lettering and indexing the various ports and valves.

236. Describe a system of governing internal combustion engines.

A four-stroke cycle engine shows a mean effective pressure calculated from the indicator diagrams of 45 lb. per square inch, but misses explosion once in every 20 revolutions; if the stroke of engine = 20 inches, diameter of cylinder = 8 inches, mean revolutions per minute = 240, estimate the I.H.P. per hour.

237. Give an account of—

(1) The pressure system of injecting oil fuel into an internal combustion cylinder;

(2) And briefly of the nature, properties, and source of supply of a fuel suitable for naval propulsion.

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